S·A·E JOURNAL

Index to Volume 51

JANUARY-DECEMBER, 1943

SOCIETY OF AUTOMOTIVE ENGINEERS, 29 West 39th Street, New York, 18

Index to Transactions

Author Index

THE Index for the Transactions Section of the SAE Journal for 1943 is printed on the following pages. Page numbers in this Section are consecutive from January to December. The table below indicates the pages contained in each issue:

January	1-32	May	149-192	September	305-344
February	33-64	June	193-228	October	345-384
March	65-104	July	229-268	November	385-420
April	105-148	August	269-304	December	421-456

Following the Transactions Index there is printed the General Editorial Section Index.

(E) indicates excerpt published.

gine Lit
irect

AUTHOR INDEX

(E) indicates excerpt published.

Discussers:

Subject Index

General Editorial Section

Author Index

Abbreviations Used:

Months of the Year

January, Jn	April, Ap	July, Jl	October, Oc
February, Fe	May, My	August, Ag	November, No
March, Mr	June, Je	September, Se	December, De

(E) indicates excerpt published.
(P) indicates paper published in full.

AUTHOR INDEX

AUTHOR INDEX

AUTHOR INDEX

Author	Title of Paper	Pa
Stout, William B.	Research and Post-War	No
Stout, William B.	Future of Post-War Aviation	De
Stout, William B.	Private Flying After the War	De 31,
Stromberg, Austin W.	Engineering Design for Air Cargo System	Jn
Sullivan, H. I.	Fleet Experience with Metal Spray	Je
Swartz, Carl E.	Some Views of Modern Bearing Technology	De
Templin, Ellis W.	Reclaiming and Renewal of Worn Automotive Parts	Oc
Tichvinsky, L. M.	Corrosion of Bearing Alloys	Fe
Tincher, A. B.	Cold Welding	Je
Torrance, P. M.	Recent Progress in Synthetic Rubbers	De
Torresen, Carel T.	Standardization of Materials	Fe
Trautman, Walter C.	Hydraulic Electric Remote Controls	Je
Trautman, W. C., and R. E. Middleton	Comparison of Hydraulic and Electrical Accessory Systems in Aircraft	No
Turner, Lt.-Col. H. R.	Steel Cartridge Cases	Fe
Van Deusen, Col. E. S.	Lessons Learned from World War II About Designing for Accessibility	(P) Jl 36,
Van Zandt, J. Parker	Some Aspects of Air Cargo Operation in South America	Jn
Veal, C. B.	Cooperative Research Comes of Age	Jn 58; Fe
Wallace, Rudolph	Interchangeable Powerplants	No
Wampler, Roy W.	See to Fight Rather Than Fight to See	De
Wangelin, D. J.; R. R. Proctor, and A. J. Stock	An Instrument for Recording Blow-by Gases in an Internal Combustion Engine	De
Wanless, James F.	The Automotive Parts Situation	Ap
Warner, Edward	Speculations on Airworthiness and Operating Regulations	Oc 26,
Watson, R. A.	Bearing Materials – Present and Future	Oc
Weil, B. H.	Putting the Rubber Tree in the Shade	My
Wells, Edward C.	Some Economic Aspects of the Commercial Use of Converted Military Aircraft	De
Wetmiller, R. S., and Bruce Hegeman	Comparison of Laboratory Diesel Engine Tests with Service Performance	Jl
White, Harold S.	Specifying Surface Finishes in Gun Manufacture	Fe
Whitfield, J. E.	Hamilton-Whitfield Blower	Fe
Wiegand, F. J.	Carburetion for the Aircraft Engine	Fe
Williams, Lew	Aviation Motors	Ag
Williams, N. D.; H. L. Moir, and W. J. Backoff	The Fairbanks-Morse Diesel Engine as a Research Tool for Pre-Evaluation of Heavy Duty Motor Oils	Jl
Willis, E. J., and R. G. Anderson	Operating Temperatures and Stresses of Aluminum Aircraft-Engine Parts	My
Wilson, W. A.; Ronald B. Smith, and Alf Lysholm	The Elliott-Lysholm Supercharger	Fe
Windram, James	The Rubber Problem in the United States Today	My
Winkler, Albert H., and Emil O. Wirth	Fuel Economy and the Split Engine	No
Winslow, C. A.	Modern Filter Development as Applied to Fuel and Lube Oil Systems of Diesel Engines	Oc
Wirth, Emil O., and Albert H. Winkler	Fuel Economy and the Split Engine	No
Wolfe, Thomas	Post-War Air Traffic	Ap
Wolfe, Thomas	Post-War Commercial Aviation	Oc
Wood, Carlos	Airplane Design for Cargo Construction	Jn
Wood, Carlos	Design Considerations of the Cargo Airplane	De
Wood, John G., and R. F. Sanders	Substitute Materials – Have We Gone the Limit?	Fe
Woodhead, Harry	Manufacturer's Viewpoint on Heavy Cargo Airplanes	(E) Ap 17,
Wotherspoon, Capt. A. S.	Hitting the Target	Fe
Young, M. H., and H. H. Hanink	New Materials for Aircraft Engines	Fe
Zeder, James C.	The Activities of the War Engineering Board	Jl
Zornig, Col. H. H.	The Why of Shell Metallurgy Specifications	Fe

Subject Index

breviations Used:

Months of the Year

uary, Jn	April, Ap	July, Jl	October, Oc
ruary, Fe	May, My	August, Ag	November, No
ch, Mr	June, Je	September, Se	December, De

indicates excerpt published.
indicates paper published in full.

Page

Printed in U.S.A.

A R
M

SINCE 1915
idly until
registered in
this equipme
order to opera
produce more
a day.

One has m
complexity o
the fuel, whi
mobile engin

OF t
th
are su
most i
pressu
by th
gines
tomer

The

THI
has be
fuels
Union
in ma
a rese
Vacuu
the S
gradu

parts. Th
gasoline m
task of ide
determinin

[1]This pape
Detroit, Mich

A

S
reg
this
ord
pro
a d
C
con
the
mo

par
gas
tasl
det

I
Det

A Refiner's Viewpoint on MOTOR FUEL QUALITY

by W. M. HOLADAY
and JOHN HAPPEL
Socony-Vacuum Oil Co., Inc.

SINCE 1915 the automobile industry has developed rapidly until there are now some 32,000,000 motor vehicles registered in the United States. The major proportion of this equipment is powered by gasoline engines and, in order to operate these engines, the petroleum industry must produce more than a million and a half barrels of gasoline a day.

One has merely to lift the hood of a car to visualize the complexity of an automobile engine. The complexity of the fuel, while not so obvious, is equally great. An automobile engine may have on the average, 2000 different operation is stupendous, and only a start has been made in this direction. As fast as the results of such investigations become available, they are being applied to the manufacture of gasoline. The need for increasing quantities of higher quality fuel has resulted in the adoption of new refining processes almost as soon as they are developed in the laboratory.

It is the purpose of this paper to describe some of the processes now used in manufacturing gasoline, and to show how they are applied to make this highly specialized product.

OF the many gasoline characteristics in which the consumer is interested, a limited number are subject to close control by the refiner. The most important of these—antiknock quality, vapor pressure, and distillation—are controlled largely by the requirements of modern automotive engines as determined by road tests and by customer acceptance.

The various processes which the refiner uses to control the properties of his gasoline involve to a considerable extent the complete rebuilding of crude oil molecules which form the starting point of the refining process.

The refiner's problem includes not only the manufacture of gasoline to meet a variety of automotive requirements, but the production of sufficient quantities of fuel with the greatest economic efficiency.

★ ★ ★

THE AUTHORS: WILLIAM M. HOLADAY (M '28) has been closely identified with development work in both fuels and lubricants. He was a member of the original Uniontown Road Test Group in 1932 and has participated in many of the subsequent field activities of the SAE. He is a research engineer in the general laboratories of the Socony-Vacuum Oil Co. JOHN HAPPEL has been employed by the Socony-Vacuum Oil Co. as a chemical engineer since graduating from the Massachusetts Institute of Technology in 1930. He has worked on the technological phases of many oil refinery processes involving the manufacture of motor fuel. In recent years he has devoted special attention as well to the correlation of gasoline performance with fuel characteristics and the practical application of this information to the control of refinery operations. During the past year, he has been engaged to a major extent, in engineering problems connected with the manufacture of synthetic rubber raw materials from petroleum.

parts. The hydrocarbon compounds which compose a gasoline may amount to about the same number. The task of identifying these compounds, isolating them, and determining their effect on internal-combustion engine

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1942.]

■ Gasoline Characteristics

Before proceeding to a description of refining processes, it will be desirable first to review briefly the various properties of a good gasoline. As will be seen, not all these properties are subject to close control by the refiner. Furthermore, the refiner is faced with the problem of pro-

ducing an adequate amount of fuel at a reasonable price. However, the requirements of modern automotive engines are of major importance in the control of gasoline characteristics.

The nature of the fuel used in an engine affects performance, economy and, in many cases, the durability of mechanical parts. Judging from customer reactions and complaints, the average car owner is interested mostly in performance. Satisfactory performance includes smooth, knockless operation and full power output, freedom from vapor lock at summer temperatures, rapid warm-up in cold weather, easy starting, and good acceleration. Good economy involves a minimum consumption of fuel while retaining desirable performance characteristics. To insure low maintenance costs, freedom from gum formation and corrosive action, together with minimum tendency toward dilution of crankcase oil are important.

From the point of view of fuel formulation, control of octane number, vapor pressure, and distillation are of major importance. Quiet operation is obtained in any given engine by supplying it with a fuel of sufficiently high knock rating to avoid detonation. Freedom from vapor lock is insured by proper control of vapor pressure. Ease of starting, proper warm-up, good acceleration, and freedom from excessive crankcase dilution are controlled by distillation characteristics. These are the major items subject to refinery control.

There are also several items which pertain to the long-time reliability of a fuel and its effect on engine condition. Excessive amounts of gum in gasoline may cause such troubles as sticking valves and clogged fuel passages. A high sulfur content is undesirable in gasoline because, in the presence of moisture, the exhaust gases can produce acids which may cause corrosion. All reliable refiners produce gasolines which are well on the safe side in these respects, since the cost of satisfactory control of gum and sulfur content is, in most cases, small when compared with the other items just mentioned.

■ Fuel Economy

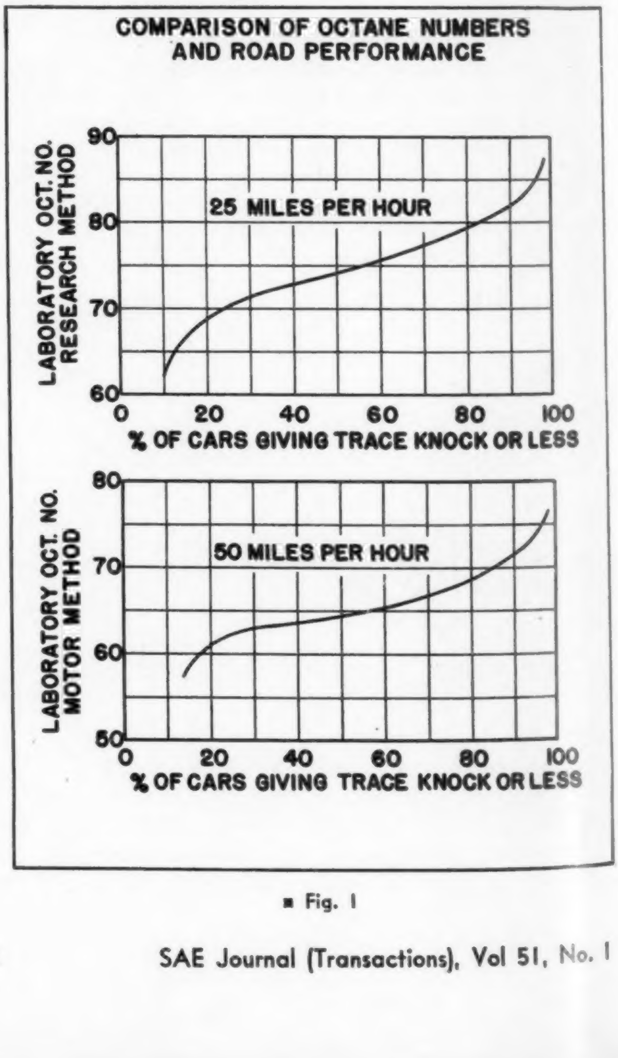
The subject of fuel economy deserves special comment. With a car performing satisfactorily in other respects, there is very little that the refiner can do about mileage. In general, fuel consumption depends on the thermal efficiency and the heat of combustion per unit weight of fuel. In the case of gasolines, variation in heat of combustion per unit weight is very small, so that the fuel consumption in miles per pound will remain practically constant. Based on this premise, a simple rule can be derived that, for fuels in the gasoline range, an increase of 3 deg API in gravity will result in a decrease in miles per gallon of about 1%. It is not possible for the refiner to exercise much control over gasoline gravity except by decreasing overall volatility or by increasing the percentage of aromatics. The use of a very low-volatility gasoline would result in the loss of starting, warm-up and acceleration performance, while the inclusion of sufficient aromatics to affect gravity appreciably is precluded by cost considerations as well as by some aspects of performance. In other words, the refiner finds that other factors than mileage control the formulation of gasoline.

Variations in fuel consumption caused by differences in driving habits, traffic conditions and, of course, by the mechanical condition of the car, are far greater than can be accounted for by fuel composition. Such effects have been covered amply by other investigators, and it is needless to discuss them at this time.

Preparations containing aniline, naphthalene, and other antiknock agents, are also sold in various attractive containers with extravagant claims of added mileage. At the prices charged for the antiknock improvement obtained, it would be far more economical to purchase a gasoline of suitable octane number as manufactured by a reliable refiner.

Antiknock Quality – Turning again to those qualities of gasoline to which the refiner must pay particular attention, one of the most important is antiknock value. Most changes in engine design today tend toward higher compression ratios with consequent improvement in efficiency and power output. In most cases, however, such changes result in increased knocking tendencies owing to the higher operating pressures thus produced. Improvements in engine efficiency have, therefore, been somewhat dependent upon the fuels of higher knock ratings.

To evaluate antiknock requirements properly, the refiner must refer to road antiknock quality and its relationship to antiknock quality as indicated by laboratory control methods on his particular fuel. The road octane number test method now in favor was adopted following tests at San Bernardino in 1940 by the Cooperative Fuel Research Committee.[1] Although insufficient data have been obtained by this method for a rigorous analysis of the influ-



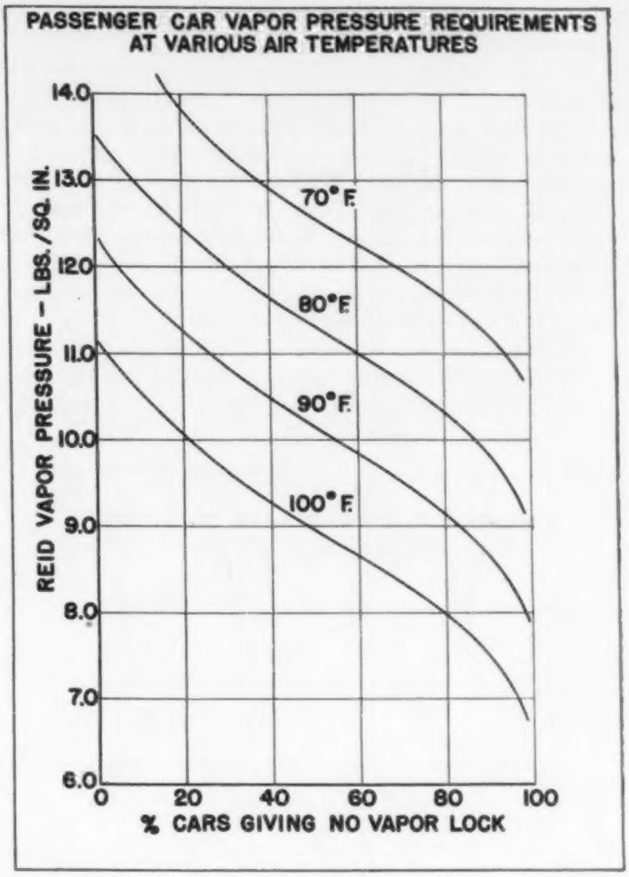
■ Fig. 1

ence of speed on knocking characteristics of all types of gasolines, the treatment of road octane values at low speed (25 mph) and a higher speed (50 mph) makes it possible to give some indication of fuel behavior under two important driving conditions. The values obtained for the 25-mph speed are indicative of knocking under the moderate conditions encountered in metropolitan and suburban traffic. Values indicated for 50 mph which, like the lower speed figures, are for full-throttle acceleration, apply to more severe driving conditions such as cross-country, hilly country, or heavy-duty commercial operation.

By combining the road octane number versus speed curves with data on passenger-car octane number requirements, it is possible to evaluate fuel ratings in terms of the percentage of cars satisfied at various speeds. Data on typical commercial gasolines would indicate that fuels having the same laboratory Research Method[2] octane number will satisfy very nearly the same percentage of cars at speeds in the neighborhood of 25 mph and fuels having the same Motor Method[3] octane number will satisfy very nearly the same percentage of cars at speeds in the neighborhood of 50 mph. Since low-speed acceleration is of predominant importance in most areas, the Research Method octane number is a very satisfactory basis for octane number specification.

Fig. 1 gives a correlation between the two laboratory octane numbers and the percentage of cars satisfied at 25 and 50 mph. This plot is intended as a guide to show how laboratory inspections may be used in a general manner to control the antiknock quality of gasoline to given performance standards. It is realized that published work of other investigators has indicated the effects of such factors as octane number distribution throughout the boiling range, hydrocarbon type as indicated by specific gravity and acid heat, and tetraethyl lead content. It is our opinion, however, that, while more elaborate correlations involving more than laboratory octane numbers, may be drawn, the refiner must still rely to a great extent, in cases of major importance, on actual road-test results.

Volatility – The other major variable subject to refinery control is dependent upon the performance factors associated with proper fuel volatility. The tests most commonly used in specifying this phase of gasoline quality are Reid vapor pressure[4] and ASTM Distillation[5]. The Reid vapor pressure is a measure of the initial tendency of the fuel to vaporize, while the distillation test is a measure of



■ Fig. 2

the fractions of various boiling ranges which compose the gasoline.

Vapor lock, which is evidenced by fuel starvation due to vapor formation in the fuel system or carburetor, can be related directly to vapor pressure. Road tests have proved this relation to be true for nearly all cars, regardless of the hydrocarbon components giving rise to the vapor pressure; that is, whether the vapor pressure is caused by propane, butane, or pentane alone or a combination of these hydrocarbons, makes little apparent difference. The selection of vapor pressure as the criterion of vapor lock has been well established by extensive work in various laboratories over a number of years.[6] Fig. 2 presents a summary of this work, the data being presented on the same basis as in the case of antiknock, so that the percentage of cars giving vapor lock under given atmospheric conditions with any specified vapor pressure can be estimated readily. It will be noted that atmospheric temperature is the determining factor in cases of vapor lock, so that this difficulty is decidedly a summer problem. The percentage of cars satisfied must be considerably greater than in the case of antiknock performance because, while knocking is, to a large extent, merely an annoyance; with severe vapor lock, the engine simply will not run.

Ease of starting, which in the past was attributed generally to the fuel, is nowadays more dependent on the factors of cranking speed, ignition maintenance, and crankcase oil viscosity. No difficulty will be experienced in cold-weather starting with practically any commercial gasoline,

[1] See SAE Transactions, Vol. 36-49, May, 1941, pp. 193-204: "1940 Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, and W. M. Holaday; see also SAE Transactions, Vol. 50, October, 1942, pp.458-464 and API Proceedings, Nov. 7, 1941: "1941 CFR Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, W. M. Holaday, and C. B. Veal.

[2] See SAE Transactions, Vol. 34, June, 1939, pp. 277-280: "CFR Research Method of Tests for Knock Characteristics of Motor Fuels," developed by the CFR Committee.

[3] See ASTM Standards on Petroleum Products and Lubricants, September, 1940, pp. 174-183: **"Standard Method of Test for Knock Characteristics of Motor Fuels," ASTM Designation D 357-40.**

[4] See ASTM Standards on Petroleum Products and Lubricants, September, 1940, pp. 252-261: "Tentative Method of Test for Vapor Pressure of Petroleum Products (Reid Method)," ASTM Designation D-323-40T.

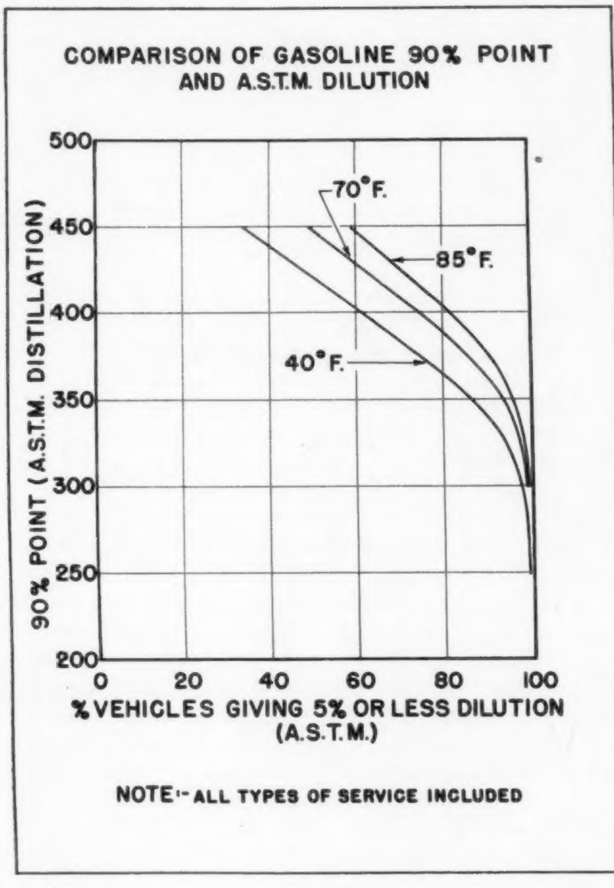
[5] See ASTM Standards on Petroleum Products and Lubricants, September, 1940, pp. 87-92: "Standard Method of Test for Distillation of Gasoline, Naphtha, Kerosene, and Similar Petroleum Products," ASTM Designation D 86-40.

[6] See SAE Transactions, Vol. 31, September, 1936, pp. 351-357: "The Correlation of Car and Fuel Vapor-Locking Characteristics," by E. M. Barber and B. A. Kulason; see also API report of Automotive Survey Committee, Dec. 15, 1938.

as nearly all of these fuels are considerably more volatile than required for this purpose.

The effect of fuel volatility on acceleration time has also been found to be very slight within the range of commercial fuels produced at the present time. It has been observed, however, that the use of more volatile fuels, such as many of the premium grades, while having no appreciable effect on acceleration time, often results in improved engine smoothness, which is noticed by many drivers. It is believed that this effect is due in part to more uniform manifold distribution with the lighter fuels, and is thus linked to the problem of satisfactory warm-up.

Dilution of the crankcase oil is caused by the entry into the crankcase of unburned gasoline. While this dilution occurs most severely in starting from cold, particularly if the engine is flooded in so doing, a certain amount of dilution is encountered under operating conditions. Other factors being equal, it has been found that dilution can be correlated reasonably well with the 90% distillation point of the gasoline, in so far as normal types of fuel are concerned. By applying this correlation to results from a large number of vehicles of different types, engaged in all types of service, Fig. 3 has been derived. It has been assumed that a maximum dilution of 5% would be within safe limits, and Fig. 3 shows the percentage of cars which would be within this value at various atmospheric temperatures with fuels of various 90% points.



■ Fig. 3

Perhaps the most critical performance characteristic which is influenced by overall fuel volatility is engine warm-up. It has been found that any fuel giving rapid warm-up will also start readily and give good acceleration. The proper formulation of warm-up specifications is thus of paramount interest to the refiner.

A considerable amount of data have been accumulated which indicates that warm-up is a function of intake manifold temperature and fuel volatility. It was found that the manifold temperature for a given car and given driving conditions rises at very nearly the same rate, regardless of the fuel used. However, the temperature required before smooth acceleration is obtained varies among different fuels, depending on their vaporizing characteristics. These characteristics, it was found, could be controlled by specifying two points on the gasoline distillation curve.

For test purposes a rather severe procedure was adopted, which consisted of starting from cold and making successive accelerations at full throttle from 10 to 20 mph in high gear, with the minimum possible use of the choke. Warm-up was considered attained when accelerations were made smoothly without the use of the choke. Results were expressed in terms of "miles to warm-up." This technique follows the procedure used by other investigators[7], although the test conditions were purposely made more severe so that "miles to warm-up" for comparable fuels are accordingly greater.

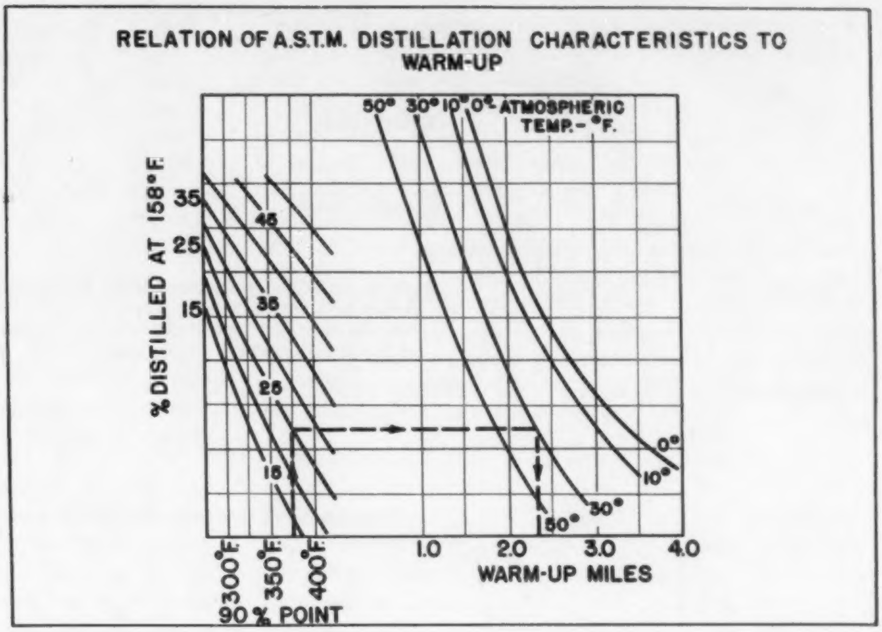
Results of this work were correlated with overall volatility expressed in terms of the percentage of fuel distilled at 158 F and the temperature at which 90% was distilled, for several atmospheric temperatures, the warm-up time being naturally affected by the initial manifold temperature. This correlation is shown in Fig. 4.

Because of the variables involved, the use of Fig. 4 is somewhat complex; hence, some explanation is in order. To find the relative warm-up performance of a fuel, start with the 90% point in the lower-left corner, proceed upwards to the per cent recovered at 158 F, across to the atmospheric temperature selected, and downwards to the scale of warm-up miles. The rate of manifold warm-up varies somewhat among car models, and the results shown on Fig. 4 represent a weighted average based on the percentage of makes on the road.

Field experience indicates that the average motorist would consider as excellent a fuel which gives warm-up in less than about 2 miles by this correlation. If, on the other hand, the warm-up is much over 4 miles, it is probable that, under normal conditions, the manifold would never reach a high enough temperature to provide smooth acceleration from 10 mph in high gear.

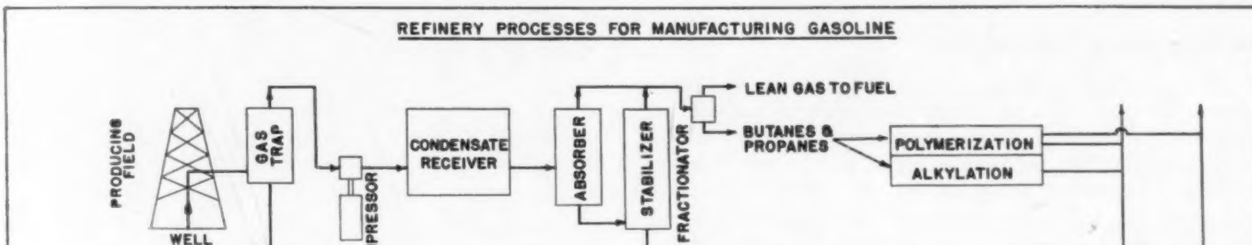
Having once established the warm-up miles desired in the fuel to be marketed in a given area, if the 90% point is fixed, it is possible to select the percentage which should be distilled at 158 F. The 90% point can be fixed from crankcase dilution considerations, previously discussed. In some cases federal, state, or municipal specifications will fix this point. It is believed that the practice of a fixed 90% point to be maintained even during warm weather is to be discouraged. From the point of view of best economy consistent with satisfactory performance, the 90% point should be varied from summer to winter as are the vapor pressure and other distillation characteristics.

[7] See SAE Transactions, Vol. 30, October, 1935, pp. 365-368: "Warming-Up Quality as a Measure of Fuel Volatility," by T. B. Rendel; see also SAE Transactions, Vol. 30, August, 1935, pp. 293-300: "A Forgotten Property of Gasoline," by J. O. Eisinger and D. P. Barnard.



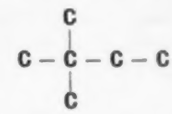
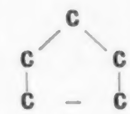
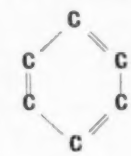
■ Fig. 4

Thus it is seen that, although there is a considerable variety of desirable performance characteristics, they are not all subject to refinery control. Certain characteristics have been standardized through years of experience and are not subject to variation by reliable refiners. Others, while important to the customer, cannot be influenced to any great extent even by an extreme variation in fuel formulation. Although there may be more elaborate and



■ Fig. 5

Table 1 – Hydrocarbon Series

Name	Type Formula	Structure	Typical Members of Series
Paraffine	$C_n H_{2n+2}$	Straight chain	C – C – C – C – C – C
Iso-Paraffine	$C_n H_{2n+2}$	Branched chain	
Olefine	$C_n H_{2n}$	Same as paraffines and iso-paraffines except each molecule contains 2 less hydrogen atoms and therefore must have a double bond between 2 of the carbon atoms	C – C – C = C – C – C
Napthene	$C_n H_{2n}$	Ring	
Aromatic	$C_n H_{2n-6}$	6-carbon ring with 3 double bonds between carbon atoms	

accurate correlations, for practical purposes the control of Research and Motor Method octane numbers, Reid vapor pressure, and ASTM distillation constitute the main considerations in the art of gasoline manufacture.

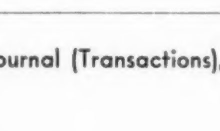
■ Petroleum Technology

In the early part of the century, the only means of producing gasoline was by simple distillation to recover the gasoline naturally occurring in crude oil. Today motor fuels are manufactured from both the higher and the lower boiling fractions of petroleum by the use of conversion processes. It has been estimated that a 90% yield of gasoline could be obtained from petroleum by using present commercial processes should the occasion demand.

To produce gasoline the manufacturer must select hydrocarbons of the proper distillation range from the crude oil, reduce the higher boiling fractions to a similar distillation range, and rearrange the molecular structure of individual hydrocarbons to the extent that such rearrangement is economical. In addition, the undesirable impurities present in the raw products must be rejected, and usually oxidation inhibitors must be added to improve stability and reduce gum-forming tendencies. Tetraethyl lead is almost always added to gasoline to improve its octane rating. Since technical knowledge on many of the operations involved is incomplete, petroleum refining is not an exact science and results are largely comparative.

A fairly good picture of the methods in use today for the manufacture of gasoline is given in the attached flow chart, Fig. 5. It should be pointed out that the arrangement shown is not necessarily the same as would be used in any given refinery but rather points out the various

Table 2 – Type Reactions in Refining Processes

Process	Charge	Product
Cracking	High Molecular Weight Hydrocarbon such as Gas, Oil or Residuum	Gasoline Range Hydrocarbons and Gas
	$C_{20}H_{42}$ →	$C_{10}H_{22} + C_{10}H_{20}$
	or	
	$C_{20}H_{42}$ →	$C_{10}H_{20} + C_9H_{18} + CH_4$
Reforming	High Molecular Weight Hydrocarbons 300 F to 450 F Boiling Pt. such as Heavy Naphtha	Gasoline Range Hydrocarbons and Gas
	Same Type Reactions as in Cracking	
Polymerization	Gaseous Olefines such as Butylene $2\,C_4H_8$ →	Gasoline Range Hydrocarbons C_8H_{16}
Alkylation	Gaseous Iso-paraffines and Olefines Iso-Butane and Butylene	Gasoline Range Iso-Paraffines Iso-Octane
		

processing possibilities. The more important refinery processes are described briefly in the following pages.

It will be realized that the chemistry of hydrocarbon technology is extremely complex. While any exposition of the subject is far beyond the scope of this paper, an attempt has been made to present, in Tables 1 and 2, a few elementary definitions and diagrams which it is hoped will help clarify the discussion of the various refinery processes.

Control of Volatility

In general, volatility control is obtained in distillation and stabilization equipment following various processing operations. Refinery conversion processes, with one or two exceptions noted later, are controlled primarily toward producing gasoline with the highest antiknock quality conducive to economical yield.

Vapor pressure is controlled in stabilizer equipment by varying the content of the highest vapor pressure components in the gasoline. These are, obviously, the lowest boiling fractions and consist almost entirely of hydrocarbons of four carbon atoms. The average vapor pressure of this mixture of butanes and butenes is about 60 psi, and the amount of this material present in gasoline is varied from about 8% in summer to some 12% in winter. In the winter the refiner is usually able to use most of the butanes and butenes which are available to him in crude oil and from conversion processes but, under summer conditions in most localities, it is often impossible to recover more than 50% in the motor fuel. The remainder must be used at lower returns as boiler and furnace fuel unless conversion equipment is at hand to process it to gasoline.

Overall fuel volatility is controlled in fractionating equipment both by varying the amount of light fractions available and by adjusting the final boiling point of products from the various refinery operations. In general, light fractions additional to those produced in ordinary crude topping and cracking operations are obtained by processing heavy gasoline fractions, purchasing natural gasoline of high volatility, and operation of gas-recovery equipment up to the maximum butane allowable by the vapor pressure specifications. For flexibility in volatility control, the refiner must rely to a large extent on varying the endpoint of the products from the various operations. The technique in endpoint control consists in cutting the various stocks to different endpoints, depending on the octane number of the base stocks. It will be noted in the following paragraphs that the straight-run stocks fall off very rapidly in octane number whereas catalytically cracked stocks show very little change in octane number as the endpoint is increased, and that thermally cracked stocks are intermediate in this respect. Therefore it is good practice to cut very low endpoints on straight-run distillates and go to higher endpoints in the various cracked stocks.

The heavy straight-run fractions may be converted to lighter material of higher antiknock value by reforming or cracking, as discussed in subsequent paragraphs.

The Control of Antiknock Quality

The refiner's controls over antiknock quality of motor fuels are:

(1) Selection of crude oil.
(2) Choice of refining process.
(3) Regulation of the addition of tetraethyl lead.

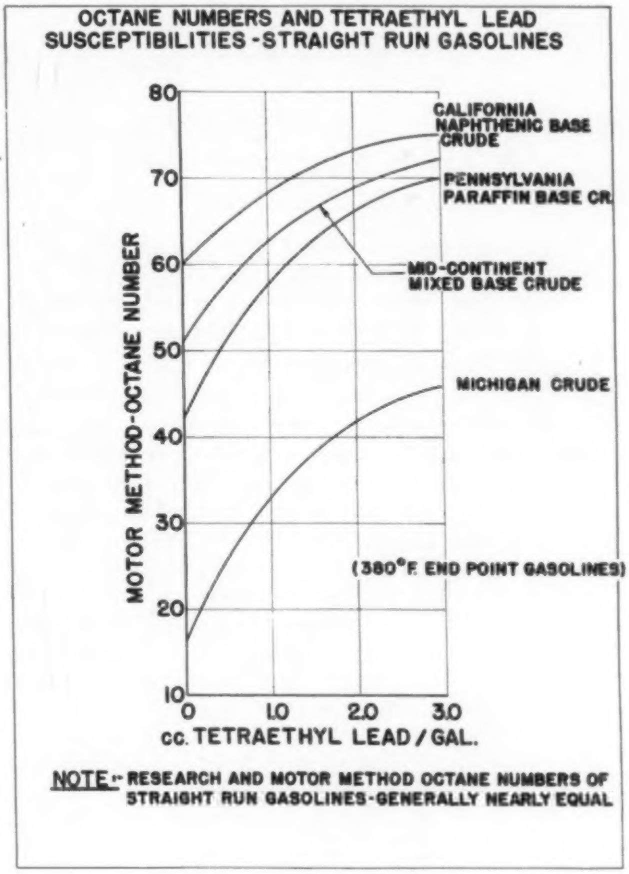
All three of these possible choices are interrelated very closely and cannot be considered individually as brought out in the following discussion.

Tetraethyl lead is added to most gasolines sold in the United States to improve the antiknock qualities. Other antiknock additives are known, but they are not widely used, mainly because of economic disadvantages.

The improvement in antiknock quality obtained with ethyl fluid varies considerably with the type of gasoline, and the economic success of a refining process may depend to a large extent upon the response of the product to tetraethyl lead. It will be pointed out in the following descriptions of conversion processes which are available to the refiner, the effectiveness of tetraethyl lead upon motor fuels obtained from those processes.

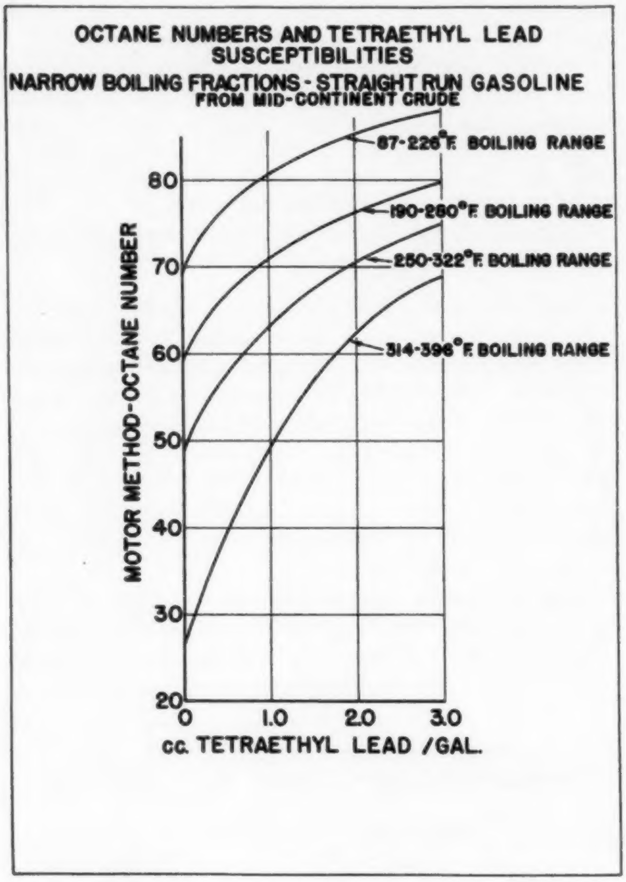
Straight-Run Gasolines – About 260,000,000 bbl per year of gasoline are produced in the United States by simple distillation of crude oil. The yield obtained in any particular case varies widely with the crude, since some crude oils contain almost no gasoline and others contain very little else. An average gasoline yield would be about 25%. The antiknock quality will also vary widely depending on the type of crude processed. Fig. 6 shows the octane number and lead susceptibility curves of four typical stocks with a 380 F endpoint.

Another significant antiknock quality characteristic of straight-run gasolines is the fact that, in most cases, octane number decreases substantially with increasing endpoint.



■ Fig. 6

Fig. 7 shows this effect in a gasoline distilled from a Midcontinent crude. General refinery procedure is to use in final blending straight-run gasolines with endpoints below 300 F. The fraction boiling in a range next above



■ Fig. 7

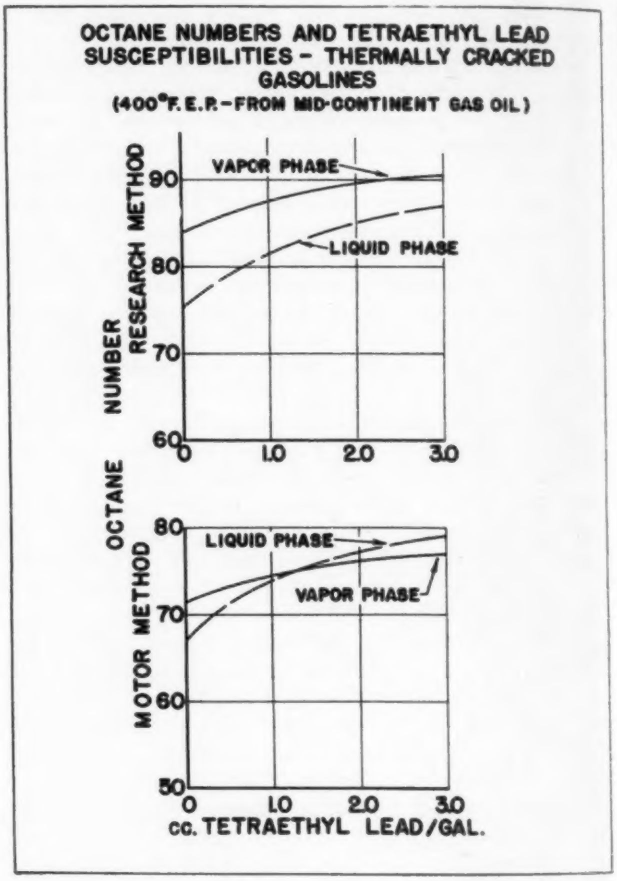
this, generally 300 F to 450 F or higher, is reformed to increase its octane number and volatility.

Research and Motor Method octane numbers of straight-run fuels are not far apart, and therefore these fuels do not show any reduction in road antiknock quality at high speeds.

Thermal Cracking – The most common process for converting high boiling petroleum constituents into gasoline is thermal cracking. This process is responsible for the production of about half of the 600 million barrels of gasoline produced in the United States annually.

Thermal cracking involves breaking down long-chain hydrocarbons above the gasoline boiling range into smaller molecules within the gasoline boiling range. The formation of non-condensible gases, heavy industrial fuel oils, and coke, occurs simultaneously. This conversion may be carried out under a wide range of high temperatures and at various pressures. These cracking conditions are dictated largely by the type of stock to be processed and by the gasoline yield and quality desired, giving due consideration to factors affecting economics, such as gas formation, coke formation, and cracking-unit maintenance. Higher temperatures and lower pressures are used in "vapor"-phase cracking than are used in "liquid"-phase cracking. The former is generally accomplished by higher gas losses and results in a more volatile gasoline of higher antiknock quality.

As just indicated, the type of charging stock greatly influences cracking conditions and characteristics of the gasoline product. Cracking stocks may vary with respect to boiling range and hydrocarbon composition. In general, charging stocks in the higher boiling ranges are cracked more easily than those in the lower boiling ranges, and therefore, do not need to be subjected to as high temperature conditions. Likewise, paraffinic charging stocks break down more readily than aromatic and naphthenic materials.



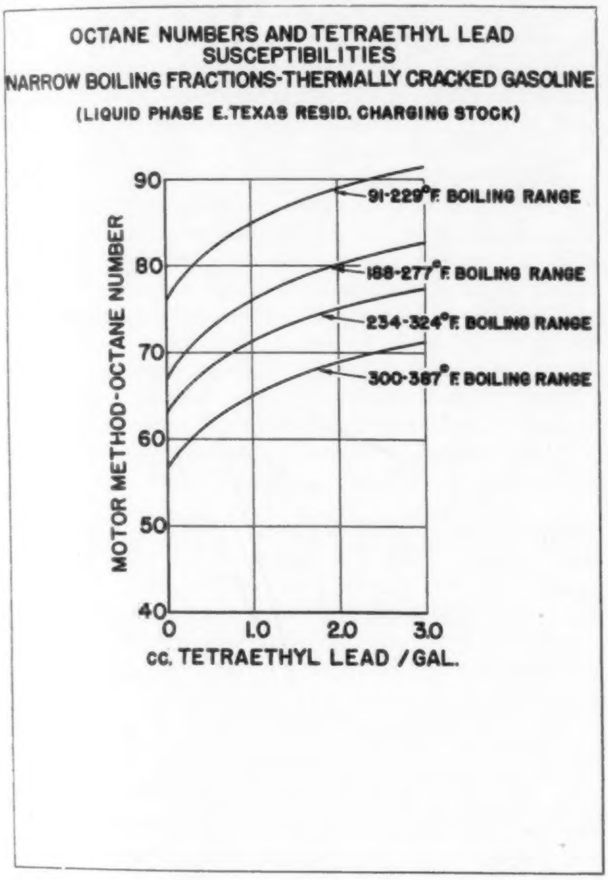
■ Fig. 8

Gasoline yields may vary widely with cracking stocks and cracking conditions, averaging about 50%.

Cracked gasolines contain a considerable proportion of olefines, which contribute particularly to their high antiknock quality. These materials, however, tend to absorb oxygen and join to one another during storage to form high boiling constituents which would not vaporize in the carbureting and manifold system of the engine, resulting in gummy deposits in the manifold and on intake valves. During recent years it has been found that this gum-forming tendency can be eliminated for storage periods of a year or more by the addition of oxidation inhibitors. As little as one part anti-oxidant in 300,000 parts of gasoline is all that is necessary. Problems arising from gum in

gasoline are now practically non-existent in automotive applications.

Fig. 8 shows the octane numbers and tetraethyl lead susceptibilities of two typical thermally cracked gasolines. It will be noted that the vapor-phase-cracked product has a higher initial octane number, but a poorer tetraethyl lead susceptibility. This is an attribute of the high olefinic hydrocarbon content of that product. Unlike straight-run gasolines, wide spreads between Research and Motor Method octane numbers exist, increasing with the severity of the cracking treatment. This characteristic, again due largely to olefinic and also aromatic hydrocarbons, is responsible for the so-called knock "sensitivity" at high speeds – the tendency for such fuels to show a decrease in road antiknock quality at high speeds. Further, as shown in Fig. 9, the spread in antiknock quality between light and heavy fractions of cracked gasolines is less than in the case of straight-run stocks.

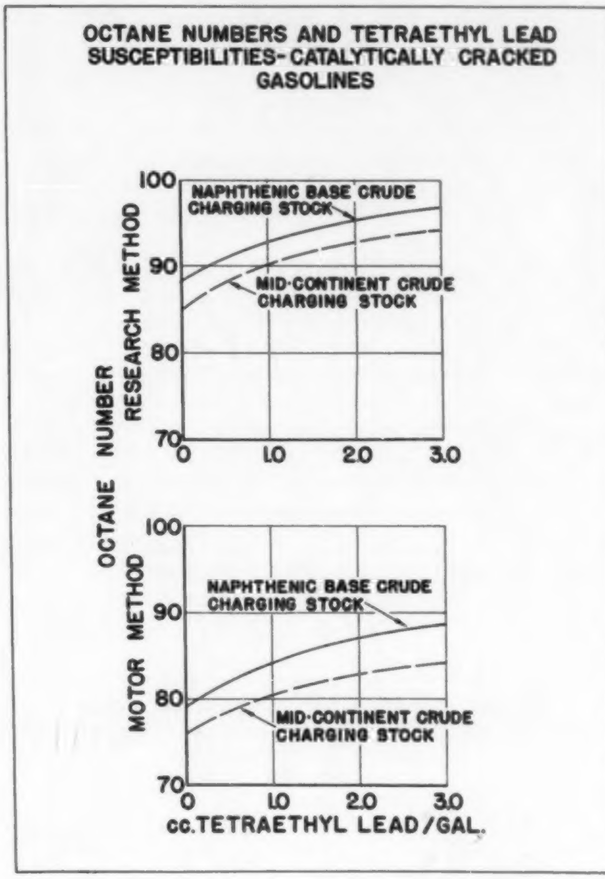


■ Fig. 9

Catalytic Cracking – About 30 million barrels of cracked gasoline are produced by the catalytic process each year, and the use of this type of operation is being extended rapidly.[8]

[8] See API Proceedings, 1938, Section III, pp. 133-148: "Catalytic Processing of Petroleum Hydrocarbons by the Houdry Process," by E. Houdry, W. F. Burt, A. E. Pew, Jr., and W. A. Peters, Jr.

As in thermal cracking, long-chain hydrocarbons above the gasoline boiling range are broken down into smaller molecules which boil within the gasoline range. However, this cracking takes place in the presence of a catalyst which serves not only to facilitate the ruptured hydrocarbon chains, but also controls the mechanism of their destruction along certain paths to form less olefinic materials and more materials of the branched-chain and aromatic types, which display better antiknock characteristics than do the olefines. Non-condensible gases and products of the domestic fuel-oil variety, generally called gas-oils, have formed simultaneously. The latter make excellent charging stocks to thermal cracking units resulting in gasolines somewhat higher in antiknock quality than those produced from virgin gas-oils.



■ Fig. 10

Catalytic cracking may be carried out from 800 to 950 F. Unlike thermal cracking low pressures, usually less than 10 psi, are used.

In general, the yield of gasoline will be about 45%, based on charge to the catalyst. The octane number (ASTM) of the gasoline produced from practically any charging stock is from 76 to 81. This is an outstanding characteristic of catalytic processing. Fig. 10 shows lead susceptibility curves of catalytically cracked gasolines from two different charging stocks. Response to lead is excellent considering the high initial octane number. Spread be-

tween Research and Motor Method rating is large, due principally to aromatic hydrocarbon content.

Fig. 11 indicates that the octane number does not change to any great extent with boiling range.

Reforming – Reforming is the term applied to the cracking of stocks in the gasoline boiling range or slightly above it. The hydrocarbon chains are rearranged under temperatures in excess of 1000 F and high pressures in the thermal process, or under milder conditions catalytically to raise the octane number and at the same time to obtain a gasoline stock with a lower boiling range, more suitable for inclusion in the finished motor fuel. As in the case of cracking, catalytic reforming gives products superior to those obtained from the thermal process. Hydroforming is the term applied to a catalytic process for reforming in the presence of hydrogen.

Yields of gasoline usually range from 80 to 90%, depending upon the severity of processing conditions. Some light gases and a small quantity of fuel oil account for the remainder of the charge. In quality, reformed gasoline is much like the cracked gasoline produced from heavier portions of the same crude. Fig. 12 shows lead susceptibilities and Motor and Research octane numbers of a thermally reformed gasoline.

Polymerization – The processes of thermal and catalytic cracking produce annually about 300 billion cu ft of gas. This gas was formerly either burned as refinery fuel or wasted, but much of it is now converted to gasoline by polymerization processes[9]. The gasoline produced in this way amounts to about 3% of the total gasoline made in the United States.

Polymerization is the reverse of cracking. In other words, it is the linking of two or more molecules to form one molecule having a longer carbon chain and a higher boiling point. The importance of such an operation as a conservation measure is obvious. In addition, the gasoline formed is generally very high in antiknock quality.

Polymerization can be effected either thermally or with the aid of a catalyst. In the thermal process, both olefines and paraffins can be utilized, the paraffins probably being dehydrogenerated under the high-temperature conditions (around 1200 F) before linking with the other olefines present. The primary products of polymerization are olefines, but various side reactions also take place, and the resulting gasoline contains aromatics, paraffins, and naphthenes as well. Because of the high olefine content, the gasoline has a relatively low tetraethyl-lead susceptibility and a marked tendency toward high-speed knock. Catalytically polymerized gasoline contains more olefines than does the thermal product, and has therefore, a somewhat higher octane number and lower lead susceptibility.

Octane numbers and lead susceptibility data for thermal and catalytically polymer gasolines are shown in Fig. 12.

Polyforming – The term "polyforming" has been applied to a recently developed process wherein thermal polymerization of gases is combined with reforming.[10] The charge to this operation usually consists of the propane and butane fractions from the other refinery units, plus heavy straight-run naphtha. The operation is carried out at from 1025 to 1125 F, and from 1000 to 2000 psi pressure. Under these conditions, cracking and polymerization proceed simultaneously along with various complex side reactions. The unreacted propane and butane are recycled so that a high yield of gasoline is obtained. The product is similar in quality to thermal polymer gasoline. A comparison of the octane numbers and lead susceptibilities of these gasolines is shown in Fig. 12.

Alkylation – Alkylation is the term applied to the reaction of an olefine with a paraffine or saturated hydrocarbon. The products consist almost entirely of members of the iso-paraffine hydrocarbon series in the gasoline boiling range. These materials have a highly branched molecular structure and are very high both in antiknock quality and in susceptibility to tetraethyl lead. In addition, they are much more stable than the gasolines produced by cracking or polymerization. This process is used to produce iso-octane from iso-butane and butylene. Neo-hexane, another fuel of very high antiknock quality, is produced by reacting ethylene and iso-butane.

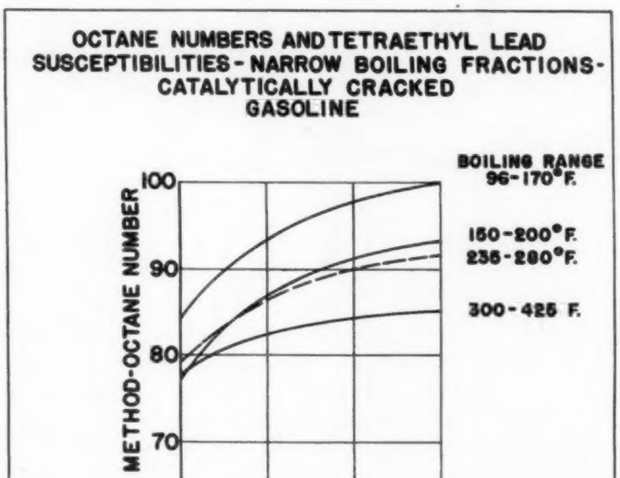
Alkylation to produce commercial iso-octane is carried out under very low temperatures and pressures using sulfuric acid as a catalyst.[11] Neo-hexane is manufactured at present by a thermal process using temperatures of 1200 F, and pressures above 3000 psi.[12] At the present time, both

[9] See API Proceedings, 1937, Section III, pp. 64-77: "Motor Fuels from Polymerization," by G. Egloff, J. C. Morrell, and E. F. Nelson.

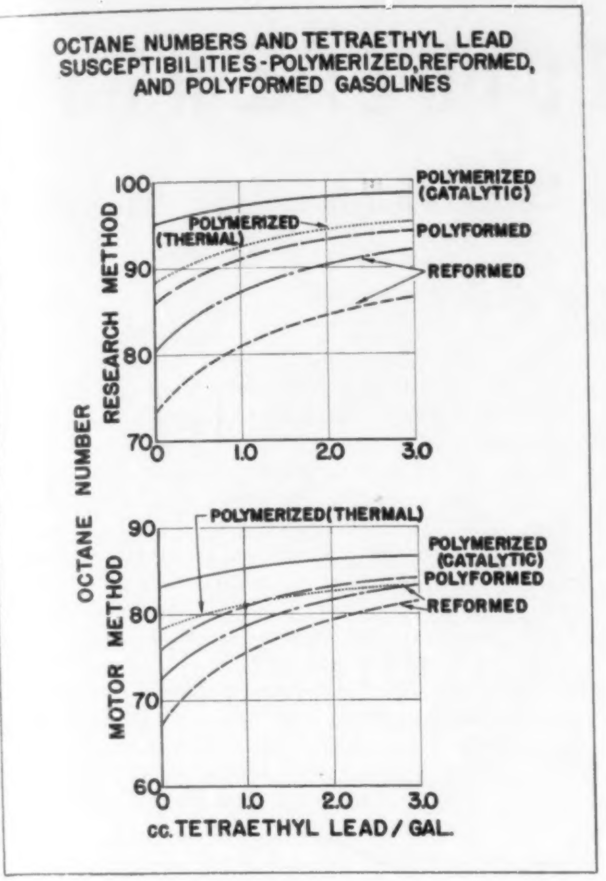
[10] See API Proceedings, 1940, Section III, pp. 17-36: "The Naphtha Polyform and Gas Reversion Processes," by J. E. Bogk, P. Ostergaard, and E. R. Smoley.

[11] See API Proceedings, 1939, Section III, pp. 89-100: "High-Octane Aviation Fuel by the Sulfuric Acid Alkylation Process," a contribution of the Anglo-Iranian Oil Co., Ltd., Humble Oil and Refining Co., Shell Development Co., Standard Oil Development Co., and The Texas Co.

[12] See API Proceedings, 1939, Section III, pp. 78-88: "Thermal Alkylation and Neo-Hexane," by G. G. Oberfell and F. E. Frey.



■ Fig. 11



■ Fig. 12

products are used exclusively for the manufacture of aviation gasoline of high antiknock quality. One-hundred octane number aviation gasoline usually contains from 40% to 50% of either of these blending agents. Present alkylate and iso-octane production amounts to about 1% of total gasoline made in the United States.[13]

For further information on individual processes, reference is suggested to The Process and Engineering Handbook, published by The Refiner.

■ Future Trends

The present war will accelerate new developments and make available fuel types which would normally require years to produce. For the duration the use of such fuels will be confined largely to military operations. The immediate effect on fuels for civilian use will be to force end-points higher and octane numbers lower in an effort to increase gasoline yield and decrease tetraethyl-lead consumption. Demands by the armored forces as well as by military aviation will result in diversion of the highest octane number stocks now available to these users.

Research activities are now stimulated by the demand for fuels of higher quality to give greater power, flexibility, and operating range to military equipment. The eventual result will probably be the formulation, at a price, of an ultimate gasoline. Even now, such a fuel is theoretically possible from purely scientific considerations. As the art of synthesizing hydrocarbon molecules advances, a gasoline could be made consisting of a blend of pure hydrocarbons, each contributing its own particular characteristics. The result would be a composite fuel of suitable volatility, high power output, and low fuel consumption for all super-power demands. Some blending agents of very high quality are already available in reasonably large quantities for aviation gasoline production. Commercial iso-octane, neo-hexane, and iso-pentane fall into this category. Certain other compounds, notably aromatics, demonstrate important advantages when burned under rich-mixture conditions as used for take-off and fighting operations with highly supercharged aviation engines.

It is beyond question that further changes in automotive engine design will be accelerated by wartime activity, and that at least some of the features of aero engine design will find application in automotive uses. Many of these changes will be for the purpose of obtaining better performance and economy from better grade fuels. It is the belief of some engineers that supercharging is the most obvious step in this direction, since the gains from further compression ratio increases appear to be limited. It would appear of equal importance to develop engine changes to take better advantage of the characteristics of current motor fuels, along the lines of better manifolding, better mixture control and better spark-advance control.

While in the future special fuels will be available for purposes where high outputs are essential and cost is not a primary consideration, it is likely that, for some time to come, the most economical powerplant combination will require an engine capable of using to the best advantage fuels of present-day quality.

■ Conclusions

It has been pointed out that, of the many gasoline characteristics in which the consumer is interested, a limited number are subject to close control by the refiner. The most important of these characteristics are antiknock quality, vapor pressure, and distillation. The various processes which the refiner uses to control the properties of gasoline involve to a considerable extent the complete rebuilding of the crude oil molecules which form the starting point of the refining process. The refiner's problem includes not only the manufacture of gasoline to meet a variety of automotive requirements but the production of sufficient quantities of fuel with the greatest economic efficiency.

The art of petroleum refining is more and more dependent on the technologist in performing this task. The ultimate in quality appears in the form of hydrocarbon molecules of highly effective characteristics. The most economical utilization of our crude oil resources consistent with the cost of the automotive equipment using the fuel will probably never call for the formulation of such gasolines except for very special purposes. It is hoped that the automotive designer will benefit from a consideration of the type of information offered herein. It is also hoped that he will be encouraged to work even more closely with the refiner and the petroleum technologist in solving their mutual problem of providing cheap power.

[13] See *The Oil and Gas Journal,* March 27, 1941, pp. 52-54: "Refiners Stand Girded for Any Emergency," by W. T. Ziegenhain.

GASOLINE ENGINE

(A Study of Odors

IT is well known that exhaust gases of bus engines, under certain conditions of operation, have a pungent odor that is disagreeable to most passengers and pedestrians. Field observations have shown definitely that this odor is present during deceleration with closed throttle and with the clutch engaged; that is, when the momentum of the bus drives the engine. It is not apparent during normal operation.

The facts – that combustion proceeds with difficulty during high-speed closed-throttle operation, and that exhaust gas during deceleration has an odor reminiscent of aldehydes, which are intermediate oxidation products – are evidence that disagreeable odor is the result of incomplete combustion.

In the present study it became necessary: to develop a simple laboratory engine test that, under controlled conditions, would produce exhaust odors similar to deceleration odors; to provide a convenient chemical substitute for the disagreeable task of smelling exhaust samples for odor intensity; and to determine the effect of certain operating variables and fuel volatility on deceleration odor.

Reference to part of this work has been made by Pardoe,[1] who subsequently also reported briefly[2] on methods used by a subcommittee of the American Transit Association on exhaust gas odors. The present paper on exhaust odors from gasoline engines follows another from this laboratory on smoke and odor from diesel engines.[3]

[This paper was presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 22, 1942.]

[1] See *Bus Transportation*, Vol. 16, 1937, pp. 480-482: "Gassing – The Bug-a-Boo of the Bus Business," by E. S. Pardoe.

[2] See American Transit Association Report, Bus Division No. 101-1, by E. S. Pardoe.

[3] See SAE Transactions, Vol. 50, December, 1942, pp. 509-520: "Effect of Diesel Fuel on Exhaust Smoke and Odor," by R. S. Wetmiller and L. E. Endsley, Jr.

THIS paper presents a study of the causes and cures of the pungent odor present in gasoline bus engines under certain conditions of operation – particularly during deceleration with the throttle closed and the clutch engaged.

To supplement the disagreeable task of smelling exhaust samples, the authors developed a chemical method for determining the odor intensity under the various operating conditions. This method was based on the discovery that formaldehyde was always present, among other aldehydes, in samples of gas that gave a characteristic odor and seemed to be its chief cause.

Since it seemed that engine speed was one of the factors contributing to the objectionable odor, closed-throttle tests were made at various speeds. It was found that the engine was using approximately the same weight of charge per suction stroke at 1500 rpm as at 600 rpm. It seemed that the residual gas dilution, that is, the ratio of residual gas in the cylinder to the fuel mixture at the time the intake valve closes, was very high at the higher speeds. Analysis showed that, at the higher speeds, combustion was much more incomplete. This is a favorable condition for the formation of aldehydes, since they are intermediate products of combustion – and, in fact, a high concentration of aldehydes was found at the upper speeds.

It was found also that large amounts of fuel in the manifold at the time the throttle is closed (and deceleration begins) cause an increase in pungency.

Using a more volatile fuel, installing higher temperature manifolds or, if the bus has been in operation a long time, cleaning the deposits from the inside walls of the manifolds, are some of the remedies suggested by the author.

■ ■ ■

THE AUTHORS: In June, 1941, J. J. MIKITA (A '36) was appointed assistant director of research of The Texas Co. Prior to that, he had been employed as mechancial engineer in the Beacon Research Laboratory of that company. Mr. Mikita received his B. S. in 1932 and his M. S. in 1933 from Pennsylvania State College. HARRY LEVIN, except for two years in the laboratory of an edible oil concern, has been with The Texas Co., where he is supervisor in charge of the analytical and testing department. Specializing in analytical chemistry, he has presented a number of papers in this field before the American Chemical Society. He received his B. S. in chemistry from Cooper-Union Institute of Technology in 1921, and his Bachelor of Laws from the New Jersey Law School in 1927. HARRY R. KICHLINE entered his present employment in the research laboratory of The Texas Co. after spending part of one year in the petroleum refining laboratory of Pennsylvania State College, his Alma Mater. He received his B. S. in chemistry from that college in 1934.

EXHAUST ODORS

Produced Under Conditions of Deceleration

by J. J. MIKITA, HARRY LEVIN, and H. R. KICHLINE

The Texas Co.

The test engines used were a 6-cyl L-head bus engine ($4\frac{1}{4}$ x $4\frac{3}{4}$ in.) and a standard variable-compression ratio CFR engine, each coupled to a d-c dynamometer. The bus engine was equipped with a standard-model updraft carburetor. The four-bowl carburetor of the CFR engine was replaced with a standard Stromberg CS-1 carburetor having a needle-valve control main jet. Each engine was provided with means for controlling the temperature and humidity of the intake air and with auxiliary equipment for measuring air-fuel ratios. The usual measurements such as spark advance, fuel consumption, temperatures, and so on, were made in a conventional manner.

■ Engine Test Procedure

Two methods were used which, for convenience, will be called part-throttle tests and deceleration tests. Part-throttle tests consisted, essentially, of operating at constant-speed and part-throttle, the air-fuel ratio being varied for each exhaust-gas sample.

Deceleration tests were made as follows: the engine was warmed up at three-quarter throttle, 1600 rpm, and 12:1 to 13:1 air-fuel ratio; when temperatures were constant, the throttle was closed and the engine was motored at the predetermined test speed. As soon as this was reached, usually 5 sec after closing the throttle, samples of exhaust for smell and aldehyde determination were taken for 1 min at 0.54 cfm. At the end of the minute deceleration period, the throttle was opened to the three-quarter position, the engine warmed up under load, and the entire procedure repeated. Fifteen such deceleration periods were repeated for one aldehyde sample.

■ Sampling Exhaust Gas

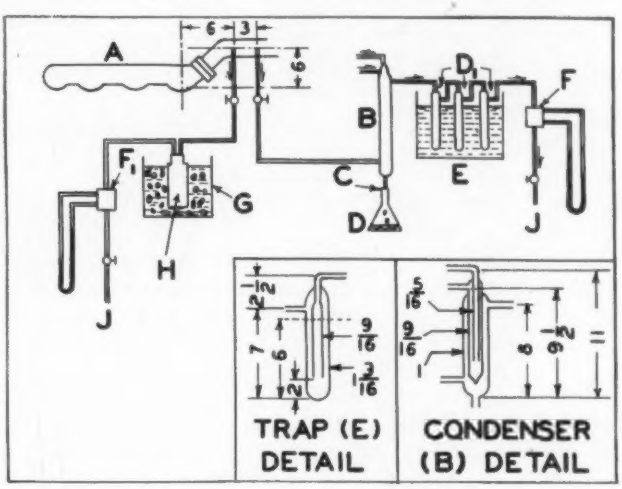
Samples for odor rating and aldehyde determination were obtained with the equipment shown in Fig. 1. Gas for aldehyde determination was taken from the exhaust pipe a short distance from the end of the manifold *A*, then passed through a water-cooled condenser *B*, three traps immersed in dry ice and chloroform *E*, and through an orifice type meter *F*. The gas was drawn at the rate of approximately 0.5 cfm for 15 1-min intervals, the condensate being ample for the test. Preliminary experiments demonstrated that the portion of exhaust gas which passes uncondensed through this train does not have the disagreeable odor. The sample taken for chemical determination of aldehydes was the combined condensates from the condenser and traps, being blended after rising to approximately the room temperature.

Samples for odor intensity rating by smell were taken from the exhaust pipe by passing the gas through a quart bottle *H* immersed in cracked ice and water *G* and through an orifice meter F_1, the gas and condensate remaining in the bottle being used.

■ Determining Intensity by Smell

The quart samples were rated in the following manner: Four or five of them were given to an individual to be rated for intensity as 1, 2, 3, and so on, 5 being the most intense and 1 the least. These samples were then given to four other individuals for rating. The accepted nasal rating was the average for the five individuals.

It is interesting to note that a particular technique was necessary to obtain consistent intensity ratings. If the tester



■ Fig. 1 – Collection of exhaust sample (Dimensions in in.)

A Exhaust manifold
B Condenser (water-cooled)
C Gas-tight connection
D&D$_1$ Condensate sample
E Traps in bath of dry ice & chloroform (—60F,
F&F$_1$ Orifice meters
G Bath of cracked ice and water
H Sample for smell
J To vacuum line

first inhaled deeply of a very strong sample, the membranes of the nose became so irritated that further ratings were, for some time, impossible. The best procedure was to uncork a bottle, bring the nose to approximately one inch from the mouth of the bottle and inhale gently. Before smelling the next sample, a pause of approximately five seconds was advisable. In a group of four samples the most and least intense were rated readily, the intermediate two required more attention.

■ Chemical Test for Exhaust Odor

In the course of the engine tests to determine the conditions of operation affecting odor of exhaust gas, it was soon found impractical to depend on the nose for rating the necessary number of samples, and an impersonal method was sought. Preliminary analyses showed that, when a gasoline bus operates under conditions of deceleration, the pungent and disagreeable exhaust always contains aldehydes in appreciable quantity. Other substances are also found, such as lower fatty acids, oxides of nitrogen, incompletely burned fuel, and so on; but it was evident that these substances did not impart the pungent odor being investigated. Chemical analysis showed that formaldehyde was present in every sample examined; acetaldehyde rarely; acrolein was never found; and a few other aldehydes which were specifically sought were either absent or the test results obtained were so questionable as to be of no value. An attempt was therefore made to correlate aldehyde content, in terms of formaldehyde, with intensity of pungent odor in exhaust gas.

For convenience in handling, the exhaust gas was condensed, and this liquid used in the subsequent examinations. The condensates do not have as disagreeable an odor as the uncondensed exhaust gas; however, nasal ratings of samples in a group regularly placed them in the same relative order whether they were collected as gas or condensed.

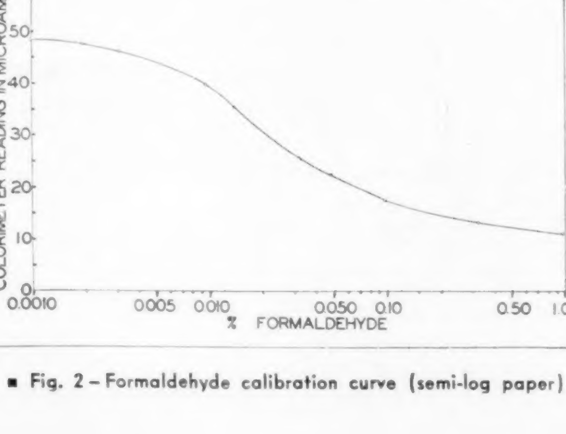
[4] See p. 1745: "Standard Methods of Chemical Analysis," by W. Scott, Fourth Edition, D. Van Nostrand Co., Inc., New York, N. Y., 1925.

[5] See the *Journal of the American Chemical Society*, Vol. 30, 1908, pp. 1607-1611: "The Colorimetric Estimation of Benzaldehyde in Almond Extracts," by A. C. Woodman and E. F. Lyford.

[6] See *Industrial and Engineering Chemistry*, Analytical Edition, Vol. 7, 1935, pp. 281-284: "A Photoelectric Colorimeter," by John H. Yoe and Thomas B. Crumpler.

[7] See p. 374: "Volumetric Analysis," by F. Sutton, Eleventh Edition, P. Blakiston's Son and Co., Philadelphia, Pa., 1924.

[8] See *Industrial and Engineering Chemistry*, Vol. 18, 1926, pp. 304-306: "Detection of Methanol in Alcoholic Beverages," by F. R. Georgia and Rita Morales.



■ Fig. 2 – Formaldehyde calibration curve (semi-log paper)

■ Determining Aldehydes in Exhaust Gas

It was convenient to employ the reaction with Schiff's[4, 5] reagent for the aldehyde determinations. The intensity of the reaction color, which is related to aldehyde content, was determined by means of a Yoe[6] photoelectric colorimeter.

Initial tests were made by measuring the time required for samples to develop a predetermined color intensity, but this proved unreliable and was not flexible enough to cover the range of samples involved. Further experiments with a series of solutions of known formaldehyde content showed good correlation between color-intensity and aldehyde content, provided a fixed time and constant temperature were used in developing the reagent-aldehyde reaction product. The actual formaldehyde content of these solutions was derived from chemical determinations on a concentrate by the method of Ripper[7], the concentrate being subsequently diluted with known amounts of water. A curve was drawn plotting micro-amperes shown by the Yoe instrument versus formaldehyde content; and, from this curve, the values of unknowns were read. Details of the procedure for determining aldehydes follow:

Reagents – Modified Schiff's Reagent[8] – Dissolve 0.2 g of Kahlbaum's rosaniline hydrochloride in 120 cc of hot water. Cool and add 2 g of anhydrous sodium sulfite dissolved in 20 cc of water, followed by 2 cc of concentrated hydrochloric acid. Dilute to 200 cc and store in a glass-stoppered amber bottle.

Apparatus –

Constant-temperature bath at 100 F
Test tubes, ¾ x 6 in.
Photoelectric colorimeter (Yoe)
Pipettes, 2 ml
Pipette, 10 ml, graduated in 0.1 ml

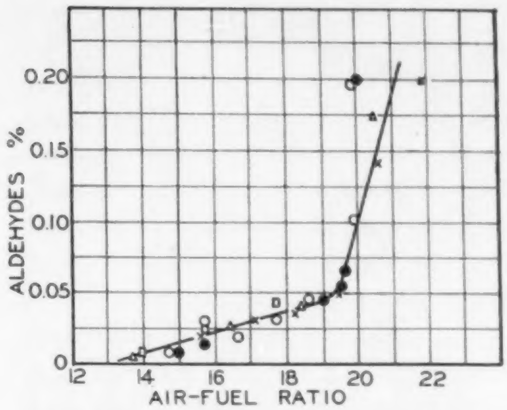
Procedure – Two ml of exhaust gas condensate are introduced into a test tube and 8 ml of Schiff's reagent added. The test tube is stoppered and placed in a bath at 100 F for 15 min to develop the reaction color. The mixture is then poured into the Yoe colorimeter tube, the test tube rinsed with 5 ml of water at 100 F, and this added to the colorimeter tube, the solution being further mixed by gentle shaking. The colorimeter tube and its contents are immediately placed in the photoelectric colorimeter and tested against a blank consisting of 13 ml of distilled water and 2 ml of sample. This type of blank was chosen because exhaust-gas condensates are frequently cloudy, and it is best to duplicate this cloud in the blank. Employing the colorimeter in the manner described by Yoe and Crumpler[6], micro-ampere readings are taken and expressed in terms of formaldehyde by reference to the calibration curve. In the standardization of our instrument the following table of values was obtained:

% Folmaldehyde (in solution under test)	Colorimeter Reading, micro-amp
0.0010	48.3
0.0019	47.6
0.0031	46.2
0.0097	39.8
0.0139	35.5
0.0323	25.8
0.0485	22.7
0.097	17.8
0.323	13.5
0.97	11.3

The data were plotted (Fig. 2) on semi-log paper to give a suitable slope to the calibration curve and better spacing in the lower concentrations where most samples fall.

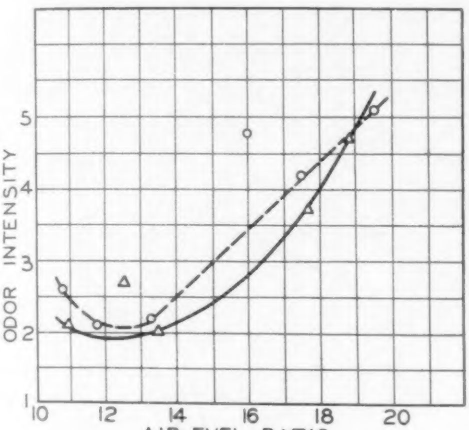
Since Schiff's reagent is rather unstable, it should be checked periodically (preferably daily) against solutions of known formaldehyde content. In duplicate tests on the same condensates, results were reproduced within "2" in the fourth decimal place on samples of low concentration.

It was found that, if a number of individuals smelled a series of four bottles of exhaust gas obtained from engines run under controlled conditions producing an assortment of intensities of pungent odor, all rated the extreme samples consistently but the order of the intermediate samples frequently was reversed. The aldehyde tests always checked the nasal ratings on extreme samples, and duplicate aldehyde tests rated intermediate samples in consistent order. This result is significant since it demonstrates that other aldehydes which may be present do not alter the correlation between odor and the chemical test based on formaldehyde. Similar correlation between smell and chemical test was obtained on solutions of known strength prepared from pure formaldehyde.



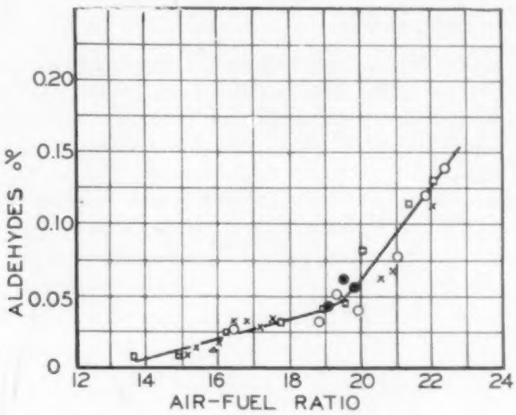
■ Fig. 5 – Effect of air-fuel ratio on aldehydes in exhaust of bus engine (part-throttle operation)

Like symbols indicate samples taken during one run. Different symbols indicate different engine runs



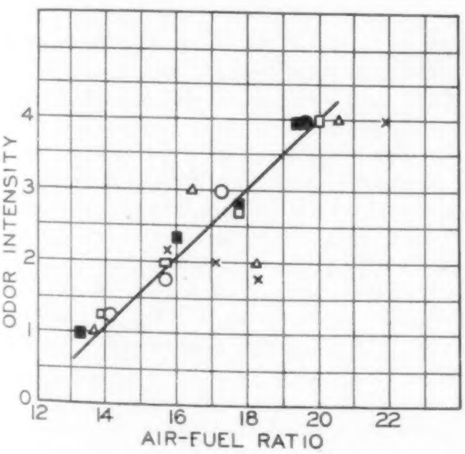
■ Fig. 3 – Effect of air-fuel ratio on exhaust odor of bus engine

○ half-throttle operation
△ quarter-throttle operation 1200 rpm



■ Fig. 6 – Effect of air-fuel ratio on aldehydes in exhaust of CFR engine (part-throttle operation)

Like symbols indicate samples taken during one run. Different symbols indicate different engine runs



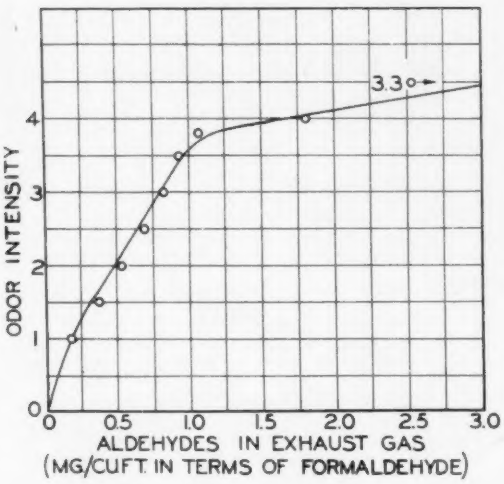
■ Fig. 4 – Effect of air-fuel ratio on exhaust odor of bus engine (quarter-throttle operation; 1200 rpm)

Like symbols indicate samples taken during one run. Different symbols indicate different engine runs

■ Effect of Air-Fuel Ratio

Preliminary investigation showed that, when operating at constant speed and part throttle, air-fuel ratio had a pronounced effect on exhaust odor. Lean mixtures gave a pungent exhaust, intensely irritating to the eyes, nose, and upper respiratory tract. Rich mixtures produced a heavy sweet odor, not nearly as objectionable as that obtained with lean mixtures. The difference in odor was so pronounced that an operator could tell by smelling the exhaust gas whether the engine was running on a lean, rich, or normal air-fuel ratio.

In Figs. 3 and 4 are presented results of tests made on the bus engine at part-throttle and 1200 rpm to determine the effect of air-fuel ratio on odor intensity. The effects were similar at both part-throttle positions. At lean ratios, the odor was particularly pungent and similar to that issuing from exhausts of buses during deceleration, the latter having been observed for numerous types of buses in city and highway operation.



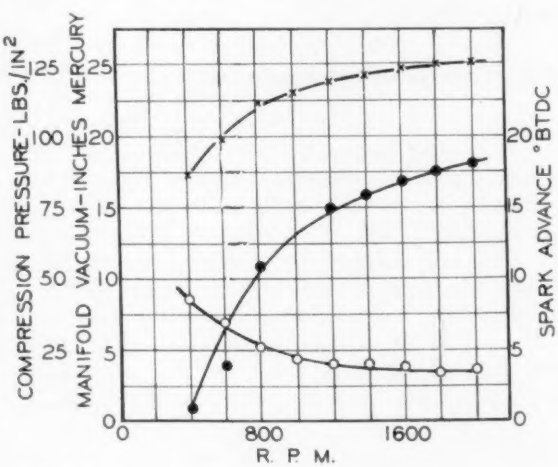
■ Fig. 7 – Odor-aldehyde correlation (exhaust gas of bus engine)

■ Operation

The effect of air-fuel ratio on aldehyde content was determined by carefully controlled tests on the bus and CFR engines, at one-quarter throttle and 1200 rpm, the air-fuel ratio being varied to obtain different intensities of odor. The relationship between aldehydes and air-fuel ratio is substantially the same for the multicylinder bus engine and the single-cylinder CFR engine, as is evident from the similarity of Figs. 5 and 6. The rather good day-to-day reproducibility of the test may be observed from these curves, results being plotted with circles for one day, squares for another, and so on.

The relationship between odor intensity and aldehyde content of exhaust gas was established by examining a number of samples taken from the bus engine at various air-fuel ratios to obtain an assortment of odor intensities. A typical curve giving this relationship is shown in Fig. 7. Other repeat tests consistently showed that intensity of odor varied as aldehyde content.

It was observed that generally an aldehyde concentration of approximately 0.75 mg per cu ft of exhaust gas marked

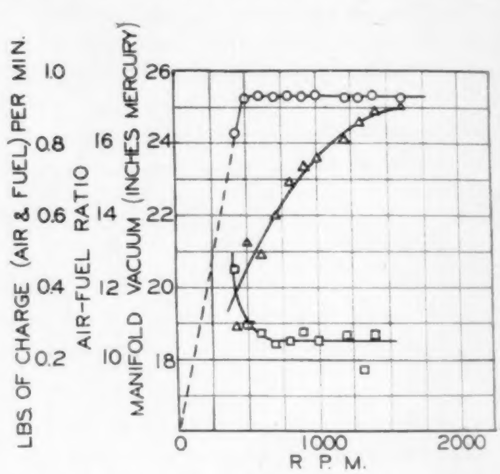


■ Fig. 8 – Effect of engine speed on compression pressure, manifold vacuum, and spark advance (bus engine; motoring with closed throttle)

○ Compression pressure
× Manifold vacuum
● Spark advance

the beginning of a very objectionable odor, causing irritation of the membranes of the eyes and nose. Above this concentration it is difficult to discriminate odor intensities by smell.

Although a sample of exhaust gas containing 0.80 mg of aldehydes per cu ft and one containing 1.50 mg per cu ft are not very dissimilar in odor, their practical effects may be vastly different. If, for example, a given volume of exhaust gas containing 0.80 mg of aldehydes per cu ft be introduced into a bus, the resulting mixture of air and exhaust gas may be below the range of objectionable odor, whereas a similar mixture with exhaust gas containing 1.50 mg of aldehydes per cu ft may fall into the objectionable class. This fact is particularly significant, for it shows the fallacy of attempting to rate exhaust odor entirely by smell, and emphasizes the advantage of an impersonal substitute such as the chemical method which has been described.



■ Fig. 9 – Effects of engine speed (bus engine; closed throttle)

○ Charge rate, lb/min
△ Manifold vacuum
□ Air-fuel ratio

■ Deceleration Tests

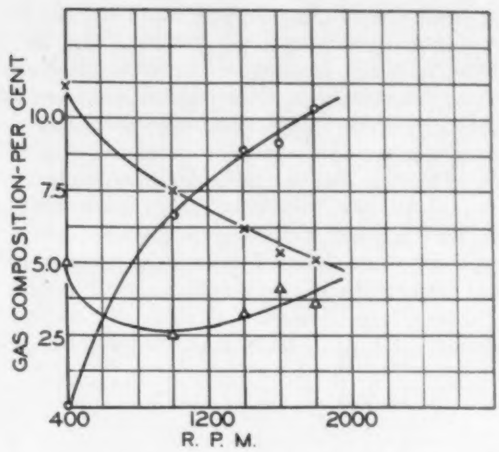
The preceding data have shown that odor and aldehydes increase as the air-fuel mixture becomes leaner. Though control of air-fuel ratio has been a great aid in the laboratory study by providing means for varying exhaust odor, it does not explain the field occurrence because it is known that buses do not operate with mixtures lean enough to cause objectionable odors. Field observations show definitely that objectionable odors occur during deceleration with closed throttle; hence operating characteristics of the bus engine under these conditions were investigated. In these tests, the engine was adjusted to idle at 400 rpm, higher speeds with closed throttle being attained by motoring with a dynamometer. In these tests, the effect of speed on compression pressures, spark advance, and manifold vacuum while motoring with closed throttle, are shown in Fig. 8. The compression pressure is almost constant between 1000 and 2000 rpm, and relatively small changes occur in manifold vacuum at speeds above 1000 rpm.

In Fig. 9 are shown the air-fuel ratio and weight of charge per minute that flows from the carburetor into the intake manifold. It should be noted particularly that, beyond 600 rpm, the charge weight and air-fuel ratio remain constant. The significance of this condition on operation is readily apparent. At 1500 rpm, the engine is operating

on the same quantity of charge as at 600 rpm, although the engine speed has increased almost three-fold. Obviously, the residual gas dilution, that is, the ratio of residual gas in the combustion chamber to the fuel mixture in the cylinder at the time the intake valve closes, is very high. To emphasize the result of closed-throttle high-speed operation, Fig. 10 has been prepared to show the weight of charge per suction stroke and the calculated resulting residual gas dilution.

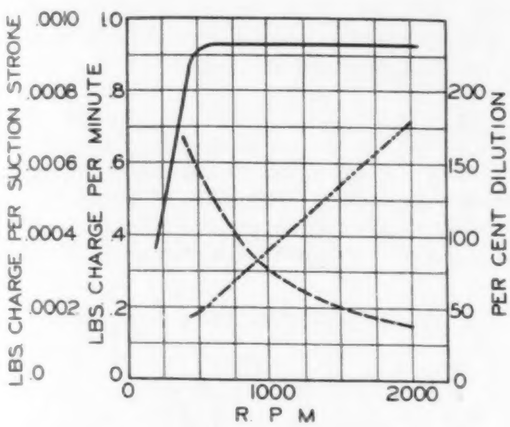
With excessive residual gas dilution, conditions for combustion are extremely adverse, and it is to be expected that odor and aldehydes will increase. That such does in fact occur was found by preliminary tests on the bus engine operated under the deceleration procedure already described. At a decelerating speed of 1600 rpm, 5 to 6 mg of aldehydes per cu ft were found in the exhaust gas, a quantity which caused an extremely pungent odor. Orsat

[9] See Philosophical Transactions of the Royal Society, Series A, Vol. 234, 1935, pp. 433-521: "Estimation of the Combustion Products from the Cylinder of the Petrol Engine and its Relation to Knock," by A. Egerton, F. Ll. Smith, and A. R. Ubbelohde.



■ Fig. 11 – Orsat analysis of exhaust gas (bus engine; deceleration with closed throttle)

O Oxygen
△ Carbon monoxide
× Carbon dioxide



■ Fig. 10 – Effect of engine speed on charge mass and residual gas dilution (bus engine; closed-throttle operation)

——— Charge rate, lb/min
— — Charge rate, lb/suction stroke
— . . Dilution

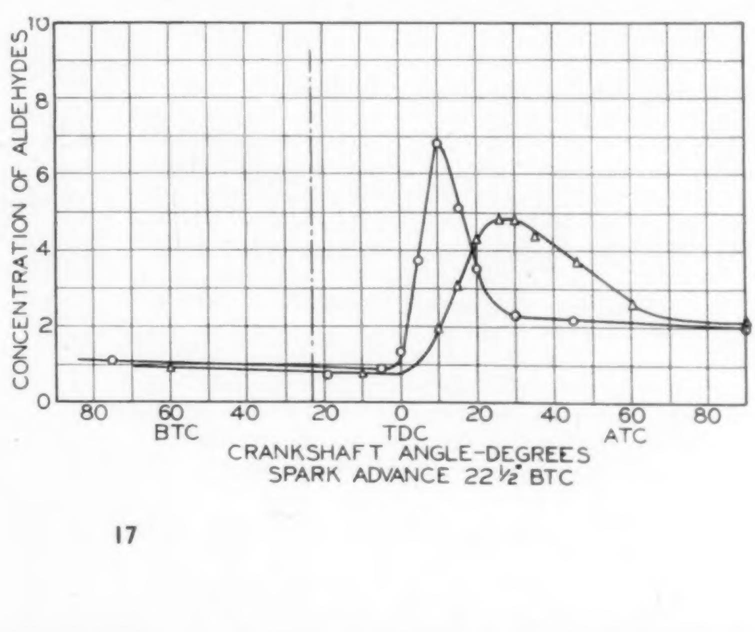
analyses showed appreciable quantities of both carbon monoxide and oxygen in the exhaust gas at higher speeds – an impossible condition if the combustion in all cylinders were normal. (Fig. 11.)

■ Cause of Deceleration Odors

From the preceding data and the work of Egerton[9], it is possible to advance an explanation for the cause of exhaust odor during deceleration. Egerton has shown that the aldehyde content of the burning charge increases to a maximum value during the early stages of combustion in an engine cylinder and, when combustion is complete, the aldehyde content is low. His results for two throttle positions are shown in Fig. 12 where it will be noted that, at the 6-deg throttle position, the aldehydes that form during combustion do so more slowly and persist longer than at 7½ deg, the larger throttle opening. Since at the smaller throttle opening residual gas dilution is greater, it may be expected that, if the latter becomes excessive and hampers combustion, the aldehyde curve may decrease at such a low rate that considerable amounts of aldehyde will be present at the time the exhaust valve opens. Thus, the quantity of aldehydes of exhaust gas depends on the completeness of combustion in the cylinder; if complete, the aldehyde

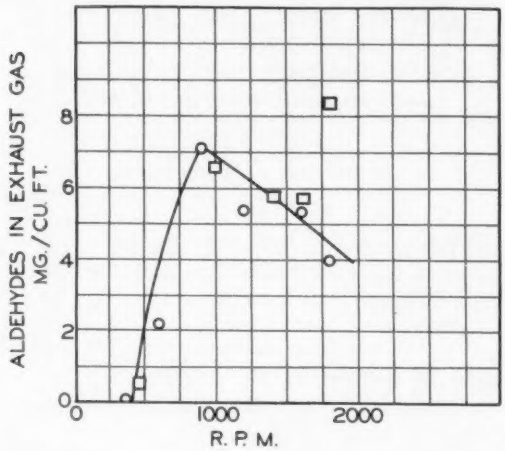
■ Fig. 12 – Formation of aldehydes during combustion (data by A. Egerton[9])

△ Throttle 6-deg open
O Throttle 7½-deg open



content is low. If burning be very slow, and the exhaust valve opens when combustion is somewhere in the maximum aldehyde stage, larger quantities of aldehydes will be found in the exhaust gas than with normal combustion. Since misfiring occurs a large percentage of the time during deceleration, it may be that large quantities of aldehydes are formed as a result of partial oxidation of the hot air-fuel mixture in those cycles where the charge is not inflamed.

With lean mixtures and consequent slow combustion, aldehydes are present in the exhaust gas because combustion was not completed in the cylinder. For a somewhat similar reason, large quantities of aldehydes are found in the exhaust gas during closed-throttle operation at high



■ Fig. 13 – Effect of decelerating speed on aldehydes in exhaust gas (bus engine)

Like symbols indicate samples taken during one run. Different symbols indicate different engine runs

speed. Though in the latter case the mixture drawn into the cylinder is a normal one, combustion is slow and incomplete because of excessive residual gas dilution, apparent in Fig. 10. The low compression pressures that exist during high-speed closed-throttle operation (Fig. 8) increase still further the tendency for incomplete combustion.

■ Speed, Idling Ratio, Spark, and Volatility

To study the effects of engine speed, idling air-fuel ratio, and spark advance on exhaust odor during closed-throttle operation at high speed, deceleration tests were made with the bus engine.

■ Engine Speed (Deceleration Tests)

The effect of decelerating speed on aldehyde content of exhaust gas is shown in Fig. 13. The quantity of aldehydes rises rapidly with speed to a maximum of about 900 rpm and slowly decreases as the speed is further increased. This behavior at higher speeds may be explained briefly by the following:

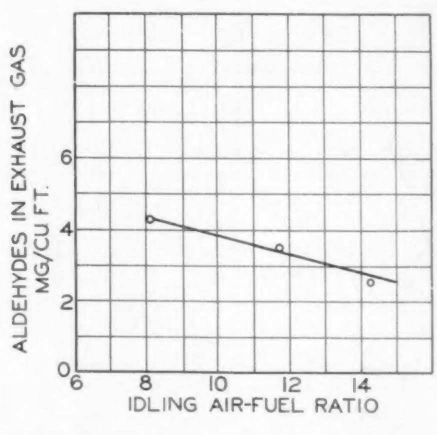
As the speed increases, residual gas dilution becomes so high that combustion will not occur. Evidence of this condition is seen in the exhaust system explosions that occur, indicating that raw fuel is being discharged from the cylinders. During the succeeding cycle, some of the residual gas and fuel are purged from the cylinder, and the fresh supply of fuel and air which is drawn in decreases the residual gas dilution. This process is repeated for several cycles until the proportion of residual gas is so low that an ignitable mixture of fuel and air is at the spark plug, and combustion is initiated. The quantity of aldehydes formed per unit volume of exhaust gas depends on the incidence of cycles in which combustion occurs; if combustion occurs in every third possible cycle, the quantity of aldehydes will be less than if it did in every cycle. Thus it appears that, as the speed is increased beyond 900 rpm and residual gas dilution becomes greater, misfiring occurs more frequently and, as a consequence, less aldehydes are formed. Cylinder purging, as described, results in loading the exhaust system with sufficient unburned fuel to cause exhaust-system explosions.

■ Idling Air-Fuel Ratio

The effect of idling air-fuel ratio is shown in Fig. 14. Rich idling mixtures produce more aldehydes during deceleration with closed throttle because the greater quantity of fuel flowing through the intake manifold will increase the fuel charge per suction stroke, decrease the number of purging cycles necessary to prepare conditions for combustion, and consequently increase the number of combustion cycles in which aldehydes are formed.

■ Spark Advance

The effect of spark advance on the aldehyde content of the exhaust gas during deceleration is shown in Fig. 15. No satisfactory explanation can be given at this time for the greater aldehyde content with increased spark advance.



■ Fig. 14 – Effect of idling air-fuel ratio on aldehydes during deceleration (bus engine)

■ Fuel Volatility

Four fuels were prepared from the same base stock and run in the CFR engine at constant speed, the air-fuel ratio being varied from 14:1 to the lean limit. The fuels had the following distillation characteristics:

	Temperature for Given % Evaporated		
Fuel	10%, F	50%, F	90%, F
A	155	226	324
B	153	261	353
C	111	218	342
D	113	259	353

The results of these tests in the CFR engine show that, at 1200 rpm with closed, one-quarter, and one-half throttle, and with intake mixture temperatures varied from 40 F to 125 F, the aldehyde content of the exhaust gas was the same for all fuels. The results of the one-quarter throttle runs are shown in Fig. 16.

Additional tests were made on two fuels in the bus engine using the deceleration procedure previously described. These fuels had the following distillation characteristics:

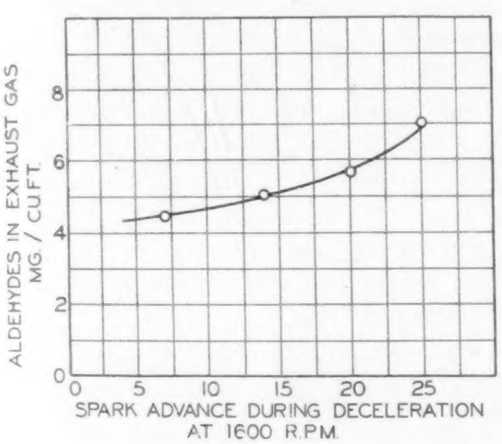
		Temperature for Given % Evaporated		
Fuel	IBP, F	10%, F	50%, F	90%, F
E	96	139	267	354
F	88	114	204	340

Tested under closed-throttle conditions at 1600 rpm and with the idling mixture ratio adjusted to a constant value, these fuels in repeat tests gave the following results:

Fuel	Aldehydes in Exhaust Gas, mg/cu ft	
E	5.4	4.4
F	3.2	3.1

which show that fuel volatility does have an effect on aldehydes during deceleration in the bus engine. Further check runs on these fuels confirmed these results.

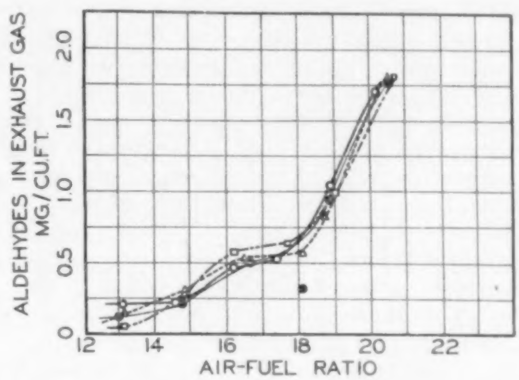
Since fuel volatility had no effect on aldehyde formation in the single-cylinder engine but did have in the multicylinder engine it, *per se,* does not influence the formation of aldehydes during combustion, and the effect produced in the multicylinder engine must be related to causes in the manifold.



■ Fig. 15 – Effect of spark advance on aldehydes in exhaust gas (bus engine)

■ Fuel Volatility and Manifolding

The amount of fuel in the manifold at the time that the throttle is closed and deceleration begins is an important variable in exhaust odor. It has been shown that enriching the idling mixture will increase odor because, with the larger quantity of fuel, less purgings of the cylinders are necessary and more aldehyde-forming combustion cycles occur. Fuel volatility plays a part in exhaust odor because it governs to some extent the amount of fuel clinging to the walls and in the pockets of the manifold. If conditions are such that large quantities of the fuel cling during normal operation, this added fuel, which is readily evaporated during deceleration (because of reduced pressure resulting from closing the throttle), furnishes the enriching charge to the cylinders to increase aldehyde formation in the same manner as previously described under "Idling Air-Fuel Ratio."



■ Fig. 16 – Effect of volatility on aldehydes in exhaust gas (CFR engine; one-fourth throttle; 1200 rpm)

○ Fuel A
△ Fuel B
□ Fuel C
● Fuel D

■ Remedies

From the nature of the causes of exhaust odor, it is obvious that no simple engine adjustment will completely eliminate it. Furthermore, it is plain that, so long as incomplete combustion occurs as a result of residual-gas dilution, changes in gasoline will not eliminate aldehydes. In exceptionally bad cases numerous expedients are available which will reduce exhaust odor to such an extent, perhaps, that noticeable improvements will be obtained. In cases of poor manifolding, a simple expedient for producing a noticeable decrease in odor is to use a very volatile fuel. Such a remedy is obtained, however, at a sacrifice because highly volatile fuels have low specific gravities and therefore result in poorer economy. Another method of producing a noticeable decrease in odor is to raise the intake mixture temperature or install higher temperature manifolds. Reports from the field indicate that, with buses that have been in operation for long periods, a satisfactory reduction in aldehydes may be obtained by cleaning the deposits from the inside of the manifolds, thus eliminating the material which traps fuel during wet mixture conditions. Adjusting the idling mixture to the leanest possible limit also reduces the aldehydes in the exhaust gas during deceleration. A mechanical device that cuts off the fuel or ignition during deceleration should be helpful.

■ Acknowledgment

The authors acknowledge their appreciation to Louis Endsley, George Raymond, and their assistants, who did a large part of the experimental work reported here, and to C. E. Cummings for helpful suggestions.

IMPORTANCE of COMPRESSION R

by MACY O. TEETOR

In Charge, Research Engineering, The Perfect Circle Co.

PISTON rings are usually classified as either compression rings to function as sealing rings or oil rings for the purpose of controlling oil consumption. Although compression rings are primarily sealing rings, they must also control some oil consumption. Oil rings prevent excessive quantities of oil from reaching the compression rings, but the amount controlled affects the sealing and lubrication of the compression rings. Therefore, it is better to consider all the rings on a piston as operating cooperatively rather than to assume that each type has a separate and distinct function independent of the others. Almost any amount of oil thrown into a cylinder can be controlled. The volume of oil to be controlled largely determines the type of oil ring to be used. For example, we can say that a plain ring used as an oil ring would be the least effective. A ventilated oil ring having narrow cylinder-contacting flanges with a spring behind it would be the most effective. Other types of oil rings would fall in between these two extremes. Their effectiveness is determined largely by design and the unit pressure they exert on the cylinder wall. The most important function of the oil ring is to allow the *correct* amount of oil to pass to the upper rings – enough to lubricate and seal the compression rings yet not more than can be controlled by the compression rings. If the oil ring does not control enough oil, the result will be excessive oil consumption and, if it controls too much oil, scuffing and excessive wear can be expected.

After a satisfactory oil ring has been selected, it is still quite a problem to select the best combination of compression rings. Different types of compression rings vary in their ability to control oil consumption. For example, we can say that a plain compression ring would control the least and a severe scraper would control the most. The amount of oil a compression ring will control can also be changed by a change in surface finish, width, radial wall thickness, unit pressure, radial wall pressure, and the distribution of this pressure around the periphery. When the oil ring is controlling about the right amount of oil, a change in compression rings may change the oil consumption of the engine several hundred per cent. If the oil ring is either controlling too much oil or allowing too much oil to reach the compression rings, the same change in compression rings might not even affect the oil consumption of the engine. Because of individual engine characteristics, the most satisfactory combination of piston rings can best be selected experimentally. Usually there are several ways to solve most of the problems, although considerable testing is sometimes necessary to finally select the best combination of rings.

★ ★ ★

"IT is better to consider all of the rings on a piston as operating cooperatively, rather than to assume that each type has a separate and distinct function independent of the others," Mr. Teetor advises in his introduction. "Because of individual engine characteristics, the most satisfactory combination of piston rings can best be selected experimentally . . . considerable testing is sometimes necessary."

Mr. Teetor also points out that:

Oil consumption can be held at a surprisingly low figure as long as scuffing is kept under control.

At excessive cylinder temperatures, compression rings lose their tension and shape characteristics and, as a result, some of their ability to follow the cylinder wall.

A piston-ring material having satisfactory heat stability does not necessarily have sufficient load-carrying capacity; cast iron, especially alloyed cast iron, still seems the best in this latter property.

THE AUTHOR: MACY O. TEETOR (M '24) has been associated with The Perfect Circle Co. since graduation from the University of Pennsylvania in 1923. After two years of engineering, he transferred to the Wharton School of Commerce and received his B.S. degree in economics. For several years after graduation he was assistant sales manager. He was then transferred to the Manufacturing Division as factory manager in charge of manufacturing. Since 1932 he has been executive engineer in charge of research.

★ ★ ★

■ Scuffing

Scuffing has been defined as the roughening of the surface of piston rings or cylinders caused by metal-to-metal contact that will smooth or heal under normal operating conditions. When we speak of adequate lubrication, we mean just enough lubrication to prevent scuffing. Compression rings must operate without scuffing on practically dry cylinder walls. Piston rings are not lubricated, that is, there is not an unbroken oil film between the ring surfaces and the cylinder wall. Sufficient oil passed by the oil ring to the compression rings to prevent metal-to-metal contact

[This paper was presented at the ASME 15th National Oil and Gas Power Conference, the SAE Diesel Engine Activity Cooperating, Peoria, Ill., June 18, 1942.]

[1] See SAE Transactions, Vol. 31, August, 1936, pp. 328-332: "Cylinder Temperature," by Macy O. Teetor.

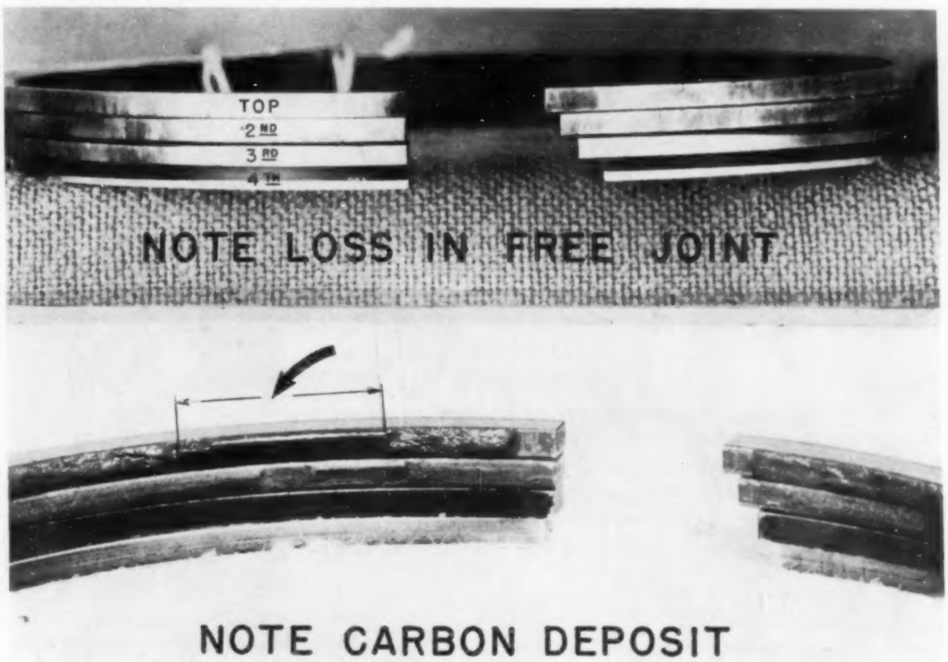
RINGS in Controlling OIL CONSUMPTION

would be such a large quantity that the oil consumption could not be controlled by any combination of compression rings known at the present time. The top compression ring is the most important ring on a piston. When scuffing is caused by piston rings, it is usually started by the top ring and oil consumption cannot be controlled satisfactorily under scuffing conditions. The top ring operates in the highest temperature zone of the cylinder, under the greatest pressure, in the part of the cylinder that has the greatest distortion, receives the least lubrication, and is blasted with any dust that enters the intake. Oil consumption can be maintained at a surprisingly low figure as long as the foregoing factors are kept under control. When they are not controlled, scuffing will start and oil consumption will become erratic or consistently excessive. The other compression rings have to operate under these same conditions, but the operating requirements decrease in intensity toward the bottom of the cylinder.

■ Temperature and Pressure

Cylinder surfaces operating at 400 F are to be expected. A temperature of 600 F in spots is not uncommon.[1] This high surface temperature can change the very thin film of oil to dry carbon. This condition is usually spotty and a new film of oil can be formed just often enough to prevent scoring, but scuffing will occur and the damage done to the rings and cylinder walls usually will be classified as excessive wear. Compression rings operating at too high a temperature produce similar results. Several conditions may be responsible for compression rings operating at a high temperature. The cylinder surface may be so hot that the heat passes from the cylinder to the rings. The rings and piston may not be dissipating the heat properly. There is a limit to the amount of heat that can be dissipated. Under scuffing conditions, the friction between the ring and the cylinder generates considerable heat, probably the most damaging heat. Regardless of where the heat comes from, compression rings lose their tension and shape characteristics at high temperatures. We have referred to the quality of a piston ring to retain its tension and shape characteristics under elevated temperatures as heat stability. The heat stability of a piston ring depends upon the ring material and the operating temperature of the ring.

When compression rings are subjected to this engine operation heat-treatment, they lose some ability to follow a cylinder wall. Then they control less oil and less blowby. The blowby passes the rings and generates more heat, burning the oil off the rings and cylinder wall which promotes scuffing. The resultant higher piston-ring temperature usually leads to ring-sticking and scoring. When rings are inspected that have been subjected to this punishment, the top ring will have lost the most free joint open-



■ Fig. 1 – Example of scuffing, loss of tension and shape characteristics of compression rings

ing, the second ring less, the third ring less, and so on down the piston. The scuffing and wear will also decrease in the same relation. It is usually difficult to tell whether the loss of tension and change in shape cause the increase in blowby which results in scuffing or whether scuffing generates the heat that causes the loss of tension and change in shape. It is probably safe to say it is a combination of both.

The pressure from the combustion chamber forces the compression rings down against the ring land on the piston and out against the cylinder wall. The usual method of increasing horsepower without increasing displacement is to increase the combustion-chamber pressure. An increase in pressure increases the load on the piston rings. This load is increased in two places. The pressure behind the rings forces the rings out against the cylinder wall with more pressure; it also forces the rings down against the ring lands on the piston with more pressure. The additional pressure on the lands increases the friction between the rings and the ring lands so that they are less inclined to slide in the grooves as the piston shifts from one side of the cylinder to the other, which transfers piston thrust from the skirt of the piston to the face of the rings. Also, cylinders that are over choked or over expanded at the top, forcing the rings to do more expanding and collapsing, transfer more load to the face of the rings as the friction is increased by additional load on the piston-ring lands.

Our present piston-ring and cylinder materials and the lubricants available are limited in their load-carrying capacity.[2] When this limit is exceeded, scuffing is sure to start. Fig. 1 is a classic example of scuffing, loss of tension and shape characteristics. The top ring lost ⅛ in. free joint opening and 1.8 lb tension, the second ring lost 3/32 in. free joint opening and 1.1 lb tension, the third ring lost 3/32 in. free joint opening and 1.2 lb tension, and the fourth ring lost 1/32 in. free joint opening and 0.4 lb tension. The wear was 0.055, 0.017, 0.012, and 0.011 in. expressed in joint increase, top, second, third and fourth ring respectively. When scuffing started, the blowby increased and the oil consumption increased, flooding the compression rings. The intense heat burned sufficient oil to carbon behind the rings which completely filled the clearance in several spots behind top ring. One point of the top ring was solidly supported by carbon so that some of the piston thrust was transferred from the piston skirt to the face of the ring. This extra load wore 0.015 in. off the face of the ring in this section in a very short time.

■ Piston-Ring Materials

These scuffing problems created by excessive temperatures and pressures may be attacked in several ways. Materials out of which piston rings can be made vary considerably in their heat stability. Cast iron is still among the best. Some alloyed cast irons have proved to be very good. Indications are that the heat stability of nitrided cast iron is very good, but other characteristics have not yet been proved satisfactory. A material having satisfactory heat stability does not necessarily have sufficient load-carrying capacity. Cast iron, especially alloyed cast iron, still seems to have the greatest load-carrying capacity. So much work is being done at the present time with different cylinder and ring materials and surface treatments that better materials should be in the offing. The selection of the best piston-ring material is complicated because the problem is essentially one of compatibility of cylinder and ring material.[3] The question of wl.at is the best piston-ring material cannot be answered definitely without taking into consideration the material of the cylinder upon which it is to operate.

Fig. 2, a picture of two pistons, illustrates several things that can be changed to improve performance. In this case it was essential that the improvement be effected with a minimum amount of experimental work. Therefore, everything was changed that had a possibility of making the operation satisfactory. The engine using the piston on the right originally produced 220 hp at 2200 rpm and consumed lubricating oil at the rate of 407 hp-hr per gal. Considerable scuffing was apparent on both rings and piston. The piston on the left reduced the oil consumption to 739 hp-hr per gal and completely eliminated the scuffing. The ring changes were as follows: cast iron to alloyed cast iron – keystone section to rectangular section – smooth surface to thread finish Ferrox surface – top, 3/16 in. SAE wall to 3/32 in. K-wall; 2nd, 3/16 in. to 1/8 in.; 3rd, 3/16 in. to 1/8 in.; 4th, slotted oil ring to the same type with more unit wall pressure; 5th, slotted oil ring to plain ring. The cylinder liners were changed from SAE 4140 steel to cast steel. The pistons were changed from raw virgin aluminum to anodized secondary aluminum. A change was also made in piston skirt shape and clearance of skirt and ring lands.

The results secured by the foregoing changes indicated that further improvement could be secured. The cylinder diameter was decreased ⅛ in.; the liners were changed to cast iron; a slotted oil ring was substituted for the plain ring in the fifth groove; and the bmep was increased. The engine still produced 220 hp at 2200 rpm with a reduction in oil consumption to 1300 hp-hr per gal. It is undoubtedly a fact that all the changes made were not necessary. In a case of this kind it is practically impossible to decide which changes were mostly responsible for the improvement. In fact, some engines have been known to show as much improvement without any *apparent* changes having been made.

■ Fig. 2 – Appearance of engine piston before and after ring, liner, and piston changes that eliminateed piston and ring scuffing and reduced oil consumption

[2] See SAE Transactions, Vol. 35, December, 1940, pp. 497-503: "Load-Carrying Capacity Phenomena of Bearing Surfaces," by Macy O. Teetor.

[3] See SAE Transactions, Vol. 33, April, 1938, pp. 137-140: "The Reduction of Piston-Ring and Cylinder Wear," by Macy O. Teetor.

CHEMICALS that AID in AIRCRAFT PRODUCTION

by RAY SANDERS
General Manager, Turco Products, Inc.

FACED with the problem of covering an extensive subject in a few pages, one must choose between full coverage of a few processes, or partial coverage of several. This paper deals briefly with several matters with the hope of generating ideas and stimulating thought on a few basic fundamentals.

RATHER than to attempt to blanket the broad field of chemicals that aid in the processing of metals in aircraft production, Mr. Sanders concentrates in this paper on a small number of protective treatments and cleaning methods and their equipment.

In a comparison of "chromatizing," phosphatizing, and anodizing, he points out that the primary purpose of these treatments is protection against corrosion, and the passivating of highly reactive metals to render a surface suitable for paint adhesion. Discussing the results of exposure tests made by the U. S. Army Air Corps to determine the relative adhesion of paints to 17ST, 24ST, and A24ST aluminum alloys after various chemical treatments and cleaning methods, he concludes that comparable results in paint adhesion will be obtained with either the anodizing, chromatizing, and phosphatizing treatments, and that these methods are well ahead of ordinary cleaning methods used before painting.

After pointing out the value of precleaning in removing the greater part of the soil, oils, and lubricants before entering the final cleaning tanks, he goes on to discuss emulsion degreasing, vapor degreasing, mechanical washing machines, and auxiliary hot tanks.

To emphasize the importance of temperature control, he reveals that the efficiency of alkaline degreasing is increased by 100% for each 20 F of temperature added above 120 F. The differences in cleaning and treating methods to be employed with aluminum and magnesium alloys are brought out, and the preparation of steel, copper, or brass for cadmium plating, and of aluminum alloys for spotwelding, is discussed at some length.

New developments discussed include a phosphoric compound which contains no aggressively active acid or virulent poison, to replace hydrofluoric acid, and a compound in jelly form for treating of sections too large to submerge in solution, to be painted over areas to be spotwelded.

THE AUTHOR: RAY SANDERS, general manager of Turco Products, Inc., has a background of 20 years in the industrial chemical manufacturing industry. For the past 10 years he has been engaged in the administration of production, distribution, and servicing of the 225 specialized industrial chemical compounds manufactured by Turco Products. In this capacity his activities have brought him into intimate contact with metal processing problems.

One interesting topic is the relative values of anodizing, chromatizing, and phosphatizing of metals as protective treatments. Perhaps the terms "chromatizing" and "phosphatizing" require explanation. They are designations that we have employed for two separate processes, and these designations have been quite universally adopted.

"Chromatizing" is meant to cover the chromic-acid treatment of aluminum alloys by the dip method. "Phosphatizing" is meant to cover the process of treating aluminum alloys with phosphoric-acid compounds. Anodizing, obviously, covers the process of building up an oxide coating on the metal by electrolysis. Each of these methods has its

[This paper was presented at a meeting of the Southern California Section of the Society, San Diego, Calif., May 8, 1942.]

place in metal processing, and each has certain advantages. The primary purpose of either anodizing, chromatizing, or phosphatizing is protection against corrosion, and the passivating of highly reactive metals to render a surface suitable for paint adhesion.

The anodizing treatment is in itself corrosion-resistant without the addition of paint while chromatizing and phosphatizing are effective corrosion resistants in combination with paint or primers.

Phosphatizing may be a desirable alternate for anodizing when paint is to be applied and where stepped-up production schedules have placed a burden on the anodizing department. Particularly useful is the W. O. No. 1 phosphate treatment for sections too large for the anodic tank and for the treatment of entirely assembled airplanes that are to be painted. Phosphatizing acts chemically with the metal to give a non-reactive passive phosphate coating that greatly improves paint adhesion and corrosion resistance.

Some exposure tests were made by the U. S. Army Air Corps to determine the relative adhesion of paints after various chemical treatments and cleaning methods. Considering all factors, the phosphatizing treatment made out very well in these tests, particularly when the simplicity of application is considered. In these exposure tests, panels of various metals were treated and painted and the panels exposed to the weather for eight months. Each panel was tested for paint adhesion after exposure for one, two, and eight months. (See Fig. 1.)

The comparisons of particular interest are those between the first, second, third, and fifth methods, as the other ordinary washing methods are generally considered as unsatisfactory.

It will be noted that, on 17ST metal, the two anodizing treatments, the phosphatizing treatment and the chromatizing treatment are all on par. The simple washing methods fall far behind, particularly for the eight months' period.

On 24ST aluminum, the sulfuric anodizing stood up better than any other treatment, although regular anodizing is close behind. The chromatizing treatment was slightly better than the phosphatizing treatment, and the washing methods far behind.

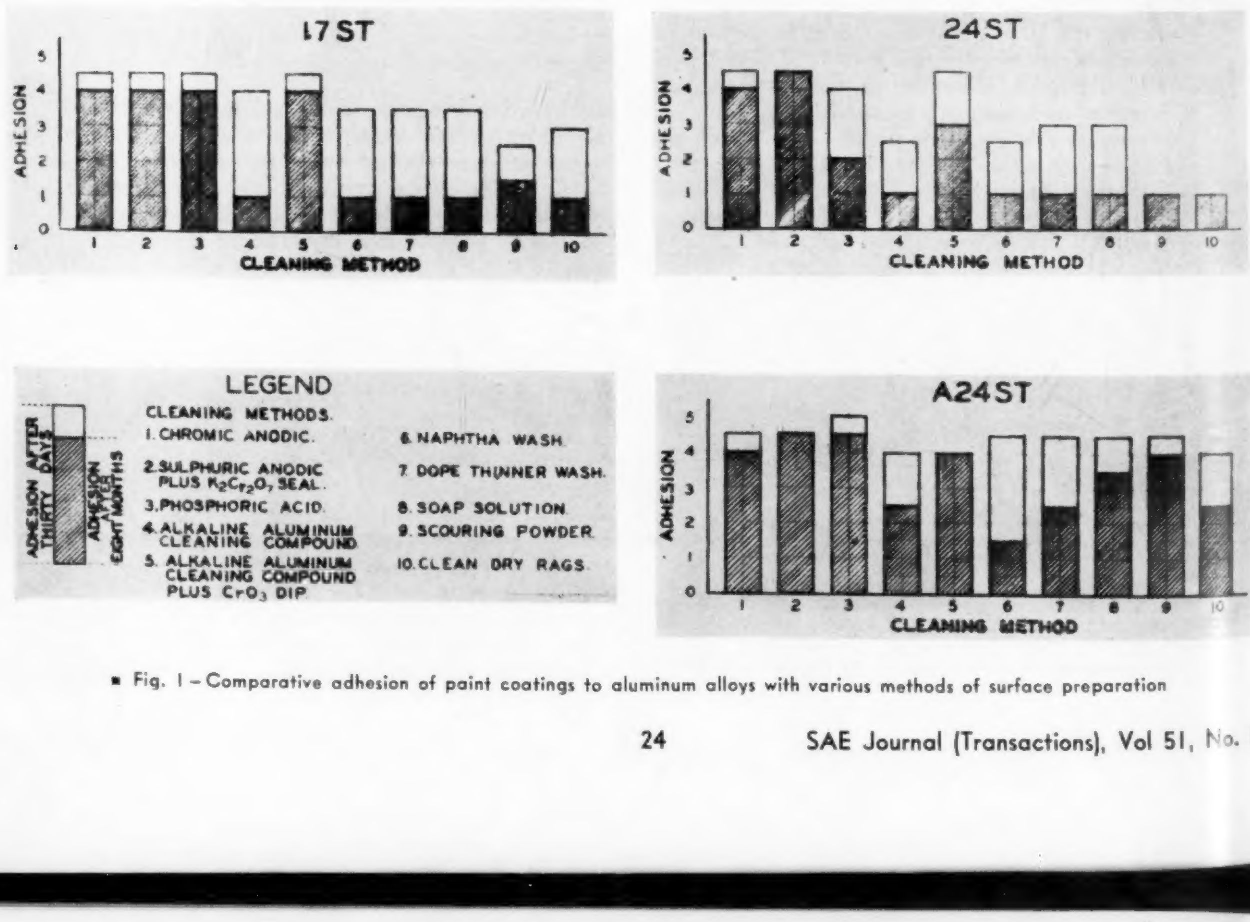
On alclad A24ST, the phosphatizing treatment was better than any other method for the one-month test and was equaled only by the sulfuric anodic process for the eight months' test.

These exposure tests also were run on several other alloys with each one showing similar variations to the three charts shown. The results of these tests would indicate definitely that, in the overall consideration, comparable results in paint adhesion will be obtained with either the anodizing, chromatizing, or phosphatizing treatments, and these are all well ahead of ordinary cleaning methods before painting.

This would appear to favor the phosphatizing method because the application of the phosphoric compound can be made to suit operating conditions and no special equipment is needed. Phosphatizing can be done by hand swabbing, a dip in ordinary dip tanks, or by spraying the surface where large areas are involved.

Usually, the required preparation of metal before anodizing, chromatizing, or phosphatizing is similar, but varies in certain essentials. In anodizing, the anodic oxide film can penetrate beneath a lightly soiled surface but, in chromatizing, the chromic acid solution will not penetrate any oil film and requires intimate contact with the metal surface. The better phosphoric solutions require less preparation as they are good penetrants and quite effective grease removers. Therefore, the highest standard of soil removal is needed before chromatizing; anodizing is slightly less critical; and in phosphatizing light soil residues can be tolerated as they will be removed by the treatment.

Proper preparation of metals for anodizing or chromatizing should include absolute removal of all soil, oil film, and residue. Frequently, much time and material are saved by the installation of a precleaning operation. The

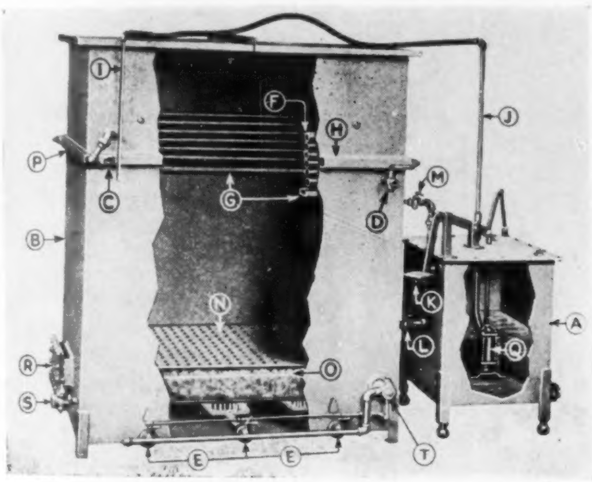


■ Fig. 1 – Comparative adhesion of paint coatings to aluminum alloys with various methods of surface preparation

term "precleaning" refers to a preliminary cleaning of metals before the final cleaning for processing. The purpose of precleaning is to remove the greater part of the more stubborn soil, such as router oils, forming lubricants, and so on, before entering the final cleaning tanks. This precleaning may be centralized in the processing department or placed as auxiliary equipment in the heat-treating, routing, or drop-hammer departments.

The possible need of a precleaning operation and the type or method to be employed is dependent on individual operating conditions. Several methods are used, including emulsion degreasing, vapor degreasing, mechanical washing machines, and auxiliary hot tanks.

Emulsion degreasing requires minimum equipment at little cost and can be done in cold solution or at low temperatures. Emulsion degreasing compounds are usually a liquid and soap-like material that is soluble in oil and also dissolves oil. Emulsion degreasing compounds are highly penetrating and potent emulsifiers. Frequently, a very short dip in a precleaner will remove alloy markings,



■ Fig. 2 – Cut-away view of a vapor degreaser

special stamping compounds, and obdurate soil that take time and are difficult to remove in alkaline hot tanks. Emulsion degreasing should be followed by a water spray rinse before the work enters the alkaline hot tanks.

The vapor degreaser type of precleaner employs a volatile liquid material. This liquid is heated to give off a vapor that condenses on the work dissolving oil and grease. Fig. 2 shows the heating elements *E*, the tank containing the liquid *O*, and the cooling coils *F*, which condense any vapors that may rise above the work. The work is hung in the tank just below the coils. Vapor degreasing is more costly to install and operate but has the advantage of requiring less space for equipment with no rinse tanks.

Mechanical washing machines (Fig. 3) have long been standard equipment in automotive and other mass-production industries. Aircraft and accessory manufacturers are adopting this method quite rapidly. This equipment usually involves special engineering for each installation and a variety of combinations of treatments may be obtained. This equipment may consist of combining submersion tanks with spray washing; use of emulsion compounds or alkaline materials or both; rinsing and drying and, in some instances, the phosphatizing treatment has also been included in the design of mechanical washing machines.

Fig. 4 shows a chemical vapor cleaner that is occasionally used as a precleaner on very large surfaces. However, this type of equipment fits into many other operations such as the removal of cosmolene from engines and ships, plant

■ Fig. 3 – Mechanical washing machine

maintenance, and so on. It combines the use of chemical solvents, pressure, water, and steam.

A cut-away view of this machine, Fig. 5, shows the solution tanks, fresh water inlet, and the pump which pumps the chemical solution and water into the coils to a restriction that builds up temperature and pressure before leaving the gun in a hot chemical vapor spray.

In some instances an additional alkaline hot tank can serve as a precleaner to conserve time and material. When

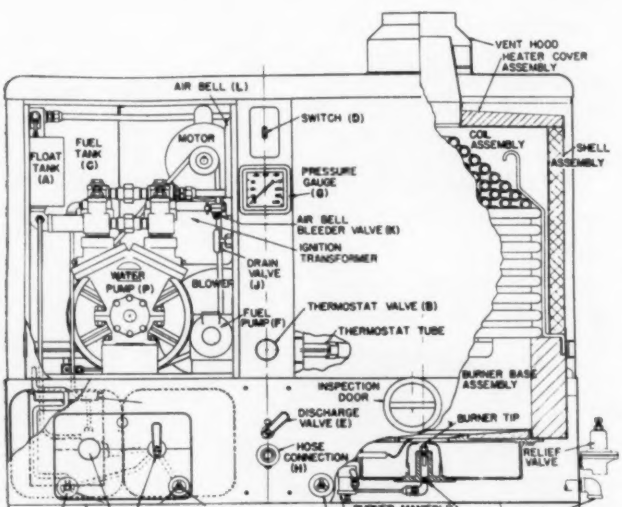
■ Fig. 4 – Chemical vapor machine

solution in the regular alkaline cleaning tank becomes soiled, this solution is transferred to the precleaner hot tank and the concentration of the solution is increased.

This stronger and soiled solution is then used to remove excessive deposits from work and prevents that soil from entering the regular hot tank. This procedure gives longer life to the regular solution and the higher concentration in the precleaner tank reduces removal time. No rinse between is required.

Irrespective of whether precleaners are used or not, a hot alkaline bath is required before anodizing or chromatizing and, unless the work is unusually clean, a hot alkaline bath should also precede the phosphatizing treatment. Chemical compounds for the hot alkaline bath should not contain any fatty acids, soap-forming ingredients, or soap. Such ingredients will combine chemically with the hardness in the water and form lime and magnesium soaps. These formations are totally insoluble and precipitate a sticky scum on the metal surface. New synthetic wetting agents are available that will act physically instead of chemically on the water hardness. Inclusion of such wetting agents in the alkaline compound formula speeds the removal operation. Rapid and free rinsing is also obtained.

The compounds for the alkaline hot bath must be inhibited properly against attack on the reactive aluminum metal. As the alkaline solution should be maintained at a



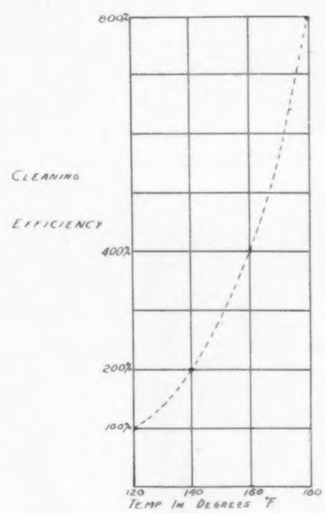
■ Fig. 5 – Cut-away view of a chemical vapor machine

pH well above 10.5, a compound of maximum alkalinity combined with absolute safety offers the best efficiency and economy.

Proper temperature control plays an important role in speed and efficiency. It is almost universally true that alkaline degreasing efficiency is increased by 100% for each 20 F of temperature added above 120 F. A solution at 160 F is 400% more effective than one at 120 F and, at 180 F, is 800% more effective. (See Fig. 6.)

Sound engineering in construction of hot tanks for alkaline degreasing is likewise important. There is much to be said on this subject, but one example is the construction to take the best advantage of convection currents. (See Fig. 7.)

Heat applied across the entire bottom of a tank causes the hot solution to rise uniformly throughout the tank, slowly and without much action. Scum or oil will float on the entire surface to be redeposited on the work when it



■ Fig. 6 – Approximate rise in cleaning efficiency due to increase in temperature

is withdrawn from the tank. By placing the heating element at the front of the tank a rolling action is obtained which gives a washing effect to the solution. Surface oil and scum are driven to the rear of the tank to avoid redisposition on the work.

In the anodizing process, the anodizing hooks, clamps, and springs must be stripped of the anodic film before re-use. Frequently this stripping is done by a hot caustic soda dip followed by a nitric acid bright dip.

The two operations can be reduced to one by adding Nitro-Brite to the nitric acid solution. The stripping is then done in one cold bath and avoids the hazardous hot caustic dip.

Magnesium metal is now being anodized at some plants, although the various Dow treatments are usually employed. The preparation of magnesium metal for treatment is different from that of aluminum. Because magnesium and aluminum are both light-weight materials and present a similar appearance, there has been some confusion regarding the chemical reaction of these two metals.

With some, the belief has existed that whatever is safe

on aluminum is also safe on magnesium. That this is not the case is illustrated by the fact that nitric acid, which is not corrosive to aluminum, etches magnesium with almost explosive violence. On the other hand, hydrofluoric acid, which dissolves aluminum, does not affect magnesium in concentrations above 10%. Mild alkaline compounds, properly inhibited against attack on aluminum, are generally not as safe on magnesium as the stronger alkaline compounds. Magnesium is affected by water and, in mild alkaline water solutions, the time of submersion should be controlled to avoid etching. Increasing the alkalinity of the solution reduces the reaction from water and increases the safety factor.

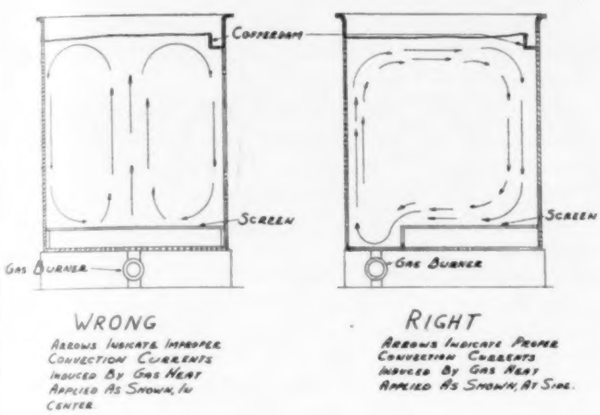


Fig. 7 – Wrong and right methods of constructing heating tanks for proper current convection

Magnesium die castings are frequently treated with cocoa butter as a mold lubricant. Cocoa butter has a high melting point and an affinity for the metal that makes removal difficult. This mold lubricant is best removed with a precleaner of the emulsion type.

Preparation of magnesium in the alkaline hot tank may be done by electrocleaning or by the straight hot alkaline bath without current.

In electrocleaning the work is made the cathode at 6 to 12 v with a current density of 10 to 30 amp per sq ft of work.

In the continuing battle against corrosion to metal, cadmium plating is used extensively for the protection of steel. Copper and brass also may be cadmium plated to prevent galvanic action.

Preparation of steel, copper, or brass for cadmium plating involves what has previously been said regarding the possible need of precleaners. In some instances precleaning is justified and in others, it is not justified.

Successful adhesion of electroplated deposits can be had only if the steel surface is absolutely chemically clean. Chemical cleanliness involves a complete removal of oil and grease, metallic oxides, mill scale, and corrosion products. Drawn shapes are usually more difficult because of the drawing compounds employed. Lubricants used for the severe forming operations are the most difficult to remove. Many of these lubricants contain sulfur in the free state but held in colloidal suspension. Sulfur is insoluble in water and quite insoluble in common organic solvents. Sulfur will react with the metal surface producing insoluble sulfides which must later be removed in the pickle bath.

Presence of such sulfides results in an extended pickling time and excessive consumption of the pickling acid. Therefore, sulfur-bearing lubricants should be avoided except where the severity of the forming operations demands them. Where possible, oils containing free fatty acids are usually favored since saponification of the acid in the alkaline hot bath assists rapid removal. If the draw is not too severe, soluble drawing oils that are emulsifiable in water are recommended.

There are two methods of operating the alkaline hot bath in the removal of soil for cadmium and other plating. One is the still tank method and the other is by electrocleaning. High temperatures are recommended and, in both instances, the alkaline bath should be kept above 180 F. The primary advantage in electrocleaning is to speed the action. The work itself is made one electrical pole, and steel electrodes are placed in the hot alkaline bath to serve as the other pole. Using the hot tank as one of the electrodes tends to cause weakening of the welds on the tank and dissipates current needlessly.

There has been considerable question whether the work should be made the anode or the cathode in the electrolytic removal of soil. This may be a matter of individual choice, but the most desirable practice has appeared to be that of first treating the work cathodically to obtain maximum gassing and then to reverse the current for a few seconds. This can be done by providing a double-throw switch. This brief anodic cleaning aids in removal of carbon smut and metallic impurities. The time of the anodic treatment must be short and insufficient for the metal to become roughened.

High conductivity of the alkaline solution itself is obviously important. An alkaline solution that builds up electrical resistance or decomposes in the electrolytic action cannot be employed. Chemical compounds with the highest conductivity are desired and, even with these compounds, it is advisable to use more concentrated solutions to improve conductivity further.

The strength of the solution should be maintained and accurately controlled to replace loss from drag-out and losses resulting from neutralization of fatty acids carried into the tank from the drawing lubricant. The average loss through drag-out on flat work is 1 gal of solution per 1000 sq ft of surface area.

The water rinse following the alkaline bath should be hot enough to keep open the pores of the metal and allow complete removal of the alkaline compound.

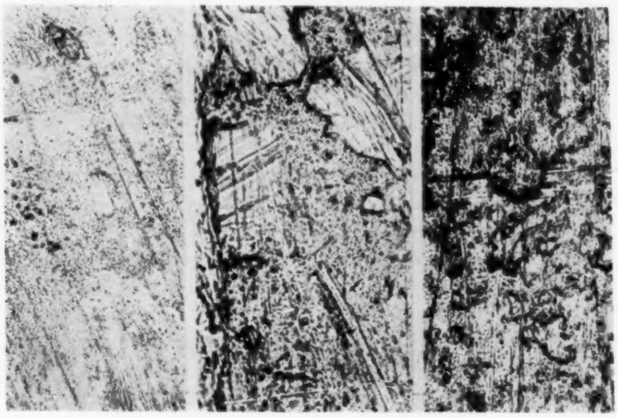
In the pickling solution preceding cadmium plating, the addition of an inhibitor will produce smooth parts free from pitting or embrittlement. The thread on screws and bolts may be impaired by uninhibited crude commercial acids. Naval Aircraft Factory Specification PT-4 requires that steel to be electroplated be pickled in acids containing an inhibitor approved under Navy Specification 51-I-2. The addition of such inhibitors to the acid solution or use of compounded pickle solutions containing approved inhibitors will reduce to 1/100th the normal attack on steel. Naturally, this addition also prolongs the life of the acid.

If the work is to be painted after cadmium plating, further treatment is advisable to provide "tooth" for the

paint or prime coat. This treatment may consist of a dip in a dilute chromic acid or a phosphatizing treatment such as Paintite.

The microphotograph, Fig. 8, shows the typical untreated cadmium-plated surface as compared with the surface after chromatizing, center microphoto, and phosphatizing, microphoto at right. These illustrations were made on the same piece of metal. The chromic acid treatment consisted of a 5% solution treated for 5 min. The phosphatized surface consisted of a 30-sec dip in a 10% solution of Turco Paintite.

Density shown in the untreated cadmium-plated surface is improved by the chromic-acid dip and substantially improved by phosphatizing. The phosphatizing treatment produces a slight etch with a fine matte to assure good paint adhesion.



■ Fig. 8 – Microphotographs, magnified 100X, showing typical untreated cadmium-plated surface (left), cadmium-plated surface after treatment with chromic acid (center), and cadmium-plated surface after phosphatizing with Paintite (right)

The Paintite phosphatizing treatment used on cadmium plating before painting is likewise used for treatment of raw steel to passivate the surface, retard corrosion, and improve paint adhesion. The phosphoric solutions employed on steel are generally of different compositions than the phosphatizing solutions previously mentioned for aluminum alloys.

In spotwelding all soil and oxides should be removed completely from the metal, and uniformity in the condition of the metal surface from weld to weld is an absolute requisite. The oxide or soil films build up electrical resistance which can, to some extent, be overcome by increasing the heat – but, if the weld should occur by building up heat to overcome this oxide resistance, the weld is still unsatisfactory. The heat will not be distributed evenly at the weld point, and any soil or oxide remaining within the weld will weaken it. This weakness may be in the form of a crack within the weld or by impairing the molecular structural affinity of the metal.

Heat is the determining element in spotwelding, and proper heat is governed by three factors. The first of these factors is the input of current through the tips and, this amount is fixed by the simple setting of the machine. The second factor governing heat is the resistance of the metal itself, and this is a fixed factor for each alloy. The third factor is the resistance occurring through interference of soil or oxides. This third and last factor governing heat is the only variable, and resistance from soil and oxides can and should be controlled by proper preparation of the metal. The welding tips also will pick up any soil or oxides on the metal to create further resistance at that point.

The microphotos, Figs. 9 and 10, illustrate the difference in results between metal properly prepared for spotwelding and uncleaned metal. These welds were made under identical operating conditions on the same metal (alclad 24ST) with the same equipment. In both instances the electrodes were cleaned and ten spotwelds made.

The metal for Fig. 9 was prepared properly for spotwelding by a method given further on in this paper, and the metal for Fig. 10 was the regular untreated stock metal. The upper portion of each picture shows a top view of the weld and the lower portion shows a microphoto of a cross-section of the same weld. The top view in Fig. 9 shows a perfectly round spot with uniform density of the metal and no indication of burning. The cross-section shows the typical shape with no cracks or flaws. The structure of the metal surrounding the weld is good, and there is an ample area of unaffected metal around the weld.

In Fig. 10 it will be observed that the top view of the spot is not round, indicating a spreading of heat and a seared condition on the metal. A slight crack appears in the center of the top view, and the dark spot would indicate that the electrode was almost immediately fouled through picking up soil or oxide.

The crack observed in the top view may also be seen in the microphoto of the cross-section below in Fig. 10 of the same weld. Additionally, the cross-section shows a void caused by gassing from the oxides or soil and a "fish mouth" appears in the edge of the weld. This weld does not provide a sufficient area of unaffected metal around the weld to give strength.

In the preparation of metal for spotwelding there are several methods available and much difference of opinion. One of the large aircraft manufacturers showed a remarkable production increase with greatly reduced man-hours by better layout and routing of work and installation of suitable cleaning compounds and equipment.

In this plant, the weekly average number of spotwelds per hour was increased from 450 spotwelds to 850 spotwelds per welder per hr. (See Fig. 11.) Simultaneously, the average number of assemblies completed per week was increased by over 34% with a reduction in man-hours of more than 38%. (See Fig. 12.)

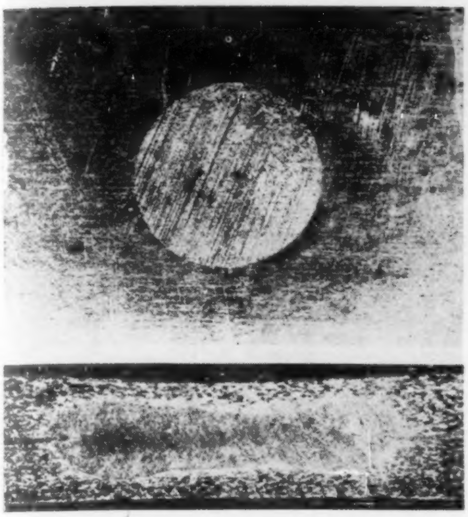
The cleaning process in this plant consists of five tanks as follows:

Tank 1 – Airlion alkaline solution
Tank 2 – Cold-water rinse
Tank 3 – 3% hydrofluoric solution
Tank 4 – Hot-water rinse
Tank 5 – Hot-air drying chamber

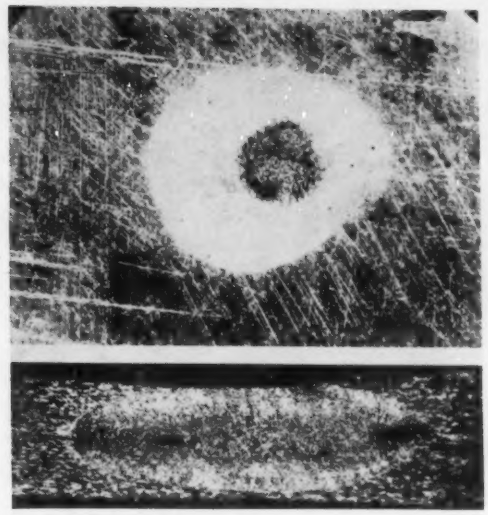
These tanks are all 3 x 17 ft x 6 ft deep. Electric timers are installed on the tanks requiring accurate control of the time factor. The flow of work is scheduled as follows:

Operation 1 – 5-min soak – Tank 1
Operation 2 – rinse – Tank 2
Operation 3 – etch – Tank 3

(Time measured by metal thickness.)



■ Fig. 9 – Spotweld – metal properly prepared



■ Fig. 10 – Spotweld – metal improperly prepared

Operation 4 – rinse – Tank 2
Operation 5 – soak – Tank 1
Operation 6 – rinse – Tank 2

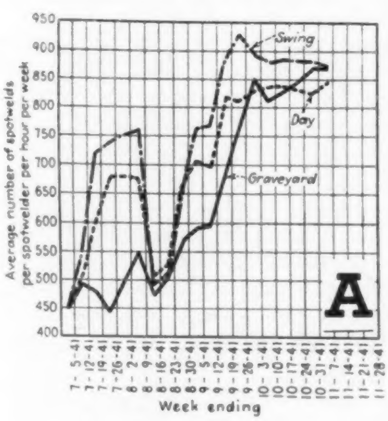
(The work is inspected and Operations 5 and 6 repeated if necessary.)

Operation 7 – rinse – Tank 4
Operation 8 – dry – Tank 5

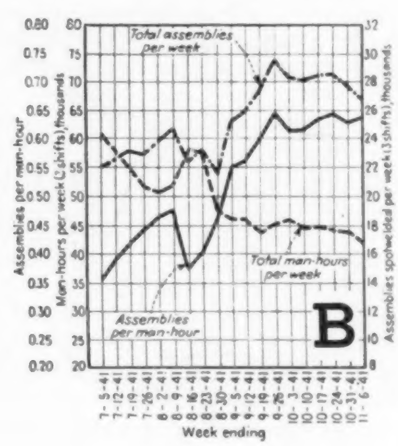
It is stated that this plant previously employed 28 men per shift in the cleaning department for spotwelding and that now twice as much work is being done with only 4 men per shift.

Some plants do not favor the use of hydrofluoric acid because of health hazards. Research has been under way to develop materials that will render the same advantages of performance as hydrofluoric acid without the hazards. Progress along this line appears encouraging with the development of the new phosphoric type of compound designated as Koldweld. This phosphoric compound contains no aggressively active acid or virulent poison. Use of this type of compound consists of a single cold dip, eliminating the need of the hot caustic etch or alkaline solution.

Another development has been a compound in jelly form for treatment of sections that are too large to sub-



■ Fig. 11 – Increase in average number of spotwelds per spotwelder per hr per week effected by better layout and routing of work and installation of suitable cleaning compounds and equipment



■ Fig. 12 – Increase in total assemblies per week and assemblies per man-hour, and decrease in total man-hours per week, following improvements in layout, routing, and cleaning methods and equipment

merge in solution. This viscous compound is painted in a stripe over the area that is to be welded. After approximately one minute, the solution is wiped off with cloths dampened with clear water, and the parts or sections spot-welded.

No discussion is given here on those other chemicals, such as paint strippers, camouflage cleaners, treatments for spray booths to break paint adhesion, stainless-steel treatments, leak detector for gas tanks and welding flux removal. These and many others form a part of the chemicals that are used in the processing of aircraft.

Wood-Plastics In Mass Production of Aircraft

THE use of wood in modern military aircraft started with the necessity of finding suitable substitutes for aluminum. The use of wood, suitably reinforced by plastic materials, will continue after the present metal shortage is relieved, because experience is showing many places where the new wood-plastic material has especially desirable characteristics. New techniques in handling the combination indicate that it can be a real mass production material.

Wood as a structural material is as old as the human race itself. Peculiarly, wood was used just as it came from the tree with little thought given to changing its characteristics until a very few years ago. Wood was the most common aircraft material until the late '20's, at which time the aircraft industry became so frightfully all-metal minded that anyone thinking in terms of wood construction was promptly classed as a has-been.

The technique of aircraft woodworking was neglected to a large extent from 1930 to 1940, except for some of the smaller ships. Most of the larger companies stopped using wood entirely. By some magic there now appears a host of aircraft woodworking experts who are prepared to build the thousands of parts which have been or must be converted to wood and plywood design. Unfortunately, many of these experts are still thinking in terms of 1928 designs and methods.

The difference between producing parts for 100 airplanes and producing parts for 10,000 airplanes requires a new trend of thought entirely. To produce parts in large quantities, it is just as important to have proper tooling for wood parts as for metal parts. It is just as important to have good production designs; just as important to have ample and satisfactory production planning. Companies which fail to realize these points cannot fit into the mass production program.

Plywood as an aircraft material has a number of structural advantages over metal, of which the major ones are:

(a) For given panel size, plywood panels are stiffer than metal panels of the same weight.

(b) For given load, plywood structures can be made with fewer stiffening members for a given skin weight.

(c) As the result of (b), plywood structures frequently can be made lighter than metal parts, yet carry equivalent loads.

(d) The greater panel stiffness in plywood parts can be used to reduce the tendency of the skin to wrinkle under load, and thereby provide better aerodynamic contours.

A few shop notes on production of wood-plastic parts seem in order.

Fixtures for mass production of wood parts must have fully as much care exercised in their design and construction as those for metal. The fixtures must be rigid. This is especially important where the fixture must carry clamping pressures. Fixtures must be properly indexed to insure complete interchangeability of parts.

For many jigs and fixtures, metal is indispensable. It is especially important that bases be made of heavy structural members to prevent distortion. It has been found that Masonite is extremely useful for many purposes.

It has been found necessary to use a hot room for accelerating the drying of glue. Most gluing is being done with cold-setting urea resin glues. These glues will set cold, as their name implies, but better joints are secured when moderate heats are used. However, the hot room is a temporary expedient, and for true production the jigs should remain fixed and local heating devices used. Several arrangements for local heating are being developed at the present time. Eventually these will be used for subassemblies, corner block attachments, and for final application of skin.

In the machining of wood parts, it has been found feasible to machine to 0.005-in. tolerances. Obviously, sawing to such limits necessitates good equipment. It cannot be done with dull blades or with loose mandrels.

Shaping is used extensively. Ribs and similar parts can be shaped to precise contours. It is even possible to produce varying bevels with proper equipment.

Routers are indispensable items of equipment. Some edge routing is done in a manner similar to shaping. Access holes and many other cutting jobs can be done conveniently with sturdy routing equipment.

Where large quantities of material of a given cross section are required, the work is most conveniently done on a sticker mill. This tool will dress four sides of the piece in one operation, and is very versatile.

■ Plywood Sheets Versus Molded Panels

Probably some mention should be made of the problem of sheet plywood construction versus molded parts. Generally, a molded part is understood to be one where the individual plies are glued together in a mold or on a mandrel in contrast to the use of plywood sheets.

In general, it is considered cheaper to use flat plywood sheets where they are applicable. This should be evident when it is realized that a single multiple-opening hot-plate press is now producing 135,000 sq ft of plywood sheets per day. It would require a great deal of molding equipment and space to produce any such amount of molded panels. The hot press mentioned above produces 32 panels in one closing.

Of course there are many places where laminating on the mold is necessary. However, some current experiments in die forming of sheet plywood are showing great promise, and it is anticipated that this will displace some of the previously molded parts where the curvature is not too great.

Excerpts from the paper of the same title by Curtis L. Bates and Harold J. Block, formerly of the Plxweve Manufacturing Co., presented at the National Aircraft Production Meeting of the Society of Automotive Engineers, Inc., at Los Angeles, Calif., Oct. 1 to 3, 1942.

PISTON RINGS and OIL CONTROL in TWO-CYCLE HIGH-OUTPUT DIESEL ENGINES

by F. GLEN SHOEMAKER and REX ALLBRIGHT

Detroit Diesel Engine Division, General Motors Corp.

THERE are two primary reasons for positive control of oil consumption in internal-combustion engines. One factor is that of oil economy, and the other is to avoid excessive deposits on the engine parts. While the matter of oil economy has enjoyed the more popular consideration, the build-up of deposits resulting from excessive oil on the hot engine parts should be given more attention. In general, the ultimate service life of the reciprocating parts in all types of engines is determined by the build-up of these deposits. Since the rate of build-up of deposits on pistons and compression rings is influenced by the quantity of oil as well as the temperature of the parts, the piston and compression rings should be lubricated sparingly.

[This paper was presented at the ASME 15th National Oil and Gas Power Conference, the SAE Diesel-Engine Activity cooperating, Peoria, Ill., June 18, 1942.]

■ Two-Cycle Valve Arrangement

The problem of oil control in the uniflow two-cycle diesel engine is different from that of the four-cycle engine, pri-

THE problem of oil control in the uniflow two-cycle diesel engine is different from that of the four-cycle engine, these authors explain, primarily because of the differences in valve arrangement. In the two-cycle valve arrangement, they point out, the oil must be controlled below the air intake ports. Any excessive oil on the portion of the piston which passes the ports will move through the ports, into the air box, and either be smeared on the compression rings or be carried into the cylinder and burned. For this reason, they explain, the oil control rings are located near the bottom of the piston so that they travel nearly to, but not past, the air intake ports.

In a concluding summary the authors list points of primary importance in the consideration of the problem of piston rings and oil control in two-cycle diesel engines.

■ ■ ■

THE AUTHORS: F. GLEN SHOEMAKER (M '20), former SAE vice-president representing the Diesel-Engine Activity, graduated from the University of Illinois in 1914 and then worked in automobile and machine shops as mechanic and manager for a number of years. He had three years as experimental engineer with the Buda Co., before spending six years at McCook Field, Dayton, in the airplane-engine laboratory of the U. S. Army Air Corps Engineering Division. This experience led to the Franklin Mfg. Co. and the design of aircooled automobile engines. For the past 10 years he has been with General Motors. When the two-cycle diesel engine program was started he was made project engineer, and has followed this development through to his present position of chief engineer of the Detroit Diesel Engine Division. REX ALLBRIGHT graduated from the University of Michigan Engineering School in 1933. During school and for a year following graduation, he was employed by the University to build apparatus for laboratory and research work in psychology. He went to the General Motors Proving Ground Section as a test engineer, Engineering Tests Division, in 1934. In 1935 he was transferred to the Mechanical Engineering Section of the Proving Ground, and later became assistant mechanical engineer. The work at the Proving Ground led to his being transferred to the Detroit Diesel Engine Division of General Motors, where he is now employed as a project engineer.

marily because of the differences in valve arrangement. In the two-cycle uniflow engine, the air enters the cylinder through ports in the cylinder wall. These ports are located so that they are uncovered by the piston as the piston approaches the bottom of its stroke.

■ Location of Oil-Control Rings

In the two-cycle valve arrangement, the oil must be controlled below the air intake ports. There must be no excessive oil on that portion of the piston skirt which passes the ports. Excessive oil here will move through the ports, into the air box, and either be smeared on the compression rings or be carried into the cylinder and burned. The oil-control rings, therefore, are located near the bottom of the piston so that they travel nearly to, but not past, the air intake ports.

The piston-pin hole must be closed off so that the oil which lubricates the piston-pin bushings cannot flow out on the piston skirt above the oil control rings.

■ Compression Rings and Piston Skirt

The piston skirt and compression rings should carry their lubricant with them as they move in the cylinder bore above the intake ports. Pistons and compression rings which have this quality make it possible to lubricate with a minimum amount of oil. This operation, in turn, results in holding the build-up of deposits to a minimum.

The surfaces of the compression ring face and piston skirt have an important bearing on their ability to carry their lubricant with them. If they are too smooth, they will not pocket and carry sufficient oil.

Another important factor in the lubrication of compression rings and pistons is the use of plated tin or other materials which have self-lubrication characteristics. If the parts suffer from lack of oil and become momentarily overheated, these materials will come into play and heal over the distressed areas.

The foregoing features of carrying oil and self-lubrication are important, not only during the break-in period, but also during the entire service life of the piston and compression rings.

■ The Oil-Control Rings

In the two-cycle diesel engine, the portion of the cylinder bore below the ports receives a generous wetting from the oil which cools the piston crown. This cooling oil is fed to the piston crown under pressure through the rifle-drilled connecting rod. The amount of oil which flows over each piston is in excess of 1 gal per min. The lower portion of the cylinder bore also receives a considerable amount of oil which is thrown off from the connecting-rod bearing.

The oil-control rings must have a high unit pressure in order to be able to control this large amount of oil. Rings with unit pressures of from 200 to 300 psi give very good results and insure positive oil control.

The passages through which the oil returns to the engine sump must be large, and they must remain clean. The three-piece oil ring (comprised of two sections and an expander) offers the opportunity of combining high unit pressure with large oil-return passages. This type of ring is also somewhat self-cleaning because of the movement of the two sections relative to each other.

Another feature of the three-piece oil ring is that it is flexible in the arrangement of its scraping hooks. The ring can be installed with both scraping hooks down to control the maximum amount of oil, or the lower section can be turned over with its scraping hook up, if it is desirable, to allow more oil to pass this ring.

Oil-control rings located in the compression ring belt have no practical value, since there is little oil to control in that area.

■ Air Flow Past the Oil Rings

Another factor which is important in controlling oil in two-cycle diesel engines which are blower-scavenged is that a small amount of air continually flows past the oil rings. In contrast to exhaust gases, this air is clean. It is drawn through the engine air cleaners by the blower and delivered to the engine air box. Since there is a large differential pressure between the air box and the crankcase, some of this air flows past the oil rings into the crankcase. The flow of air past the oil rings is of considerable value in keeping the oil passages clear. It is of benefit, also, in spreading and holding back the oil.

■ Typical Oil Consumption

Following, in tabulation form, are some typical oil consumption figures for the two-cycle, high output diesel engine. The data shown are for pistons carrying two oil-control rings. In one case both oil rings have their scraping hooks down (double-scraper position) and, in the second instance, the first ring has both hooks in the scraper position, but the second ring has its lower hook turned up (hydraulic position).

First and Second Oil Rings
Double Scraper

	1200 Rpm	2000 Rpm	2200 Rpm
Lb/Hr/Cyl	0.035	0.05	0.06
Bhp Hr/Gal	4520	5230	4560

First Oil Ring Double Scraper and
Second Oil Ring Hydraulic

	1200 Rpm	2000 Rpm	2200 Rpm
Lb/Hr/Cyl	0.055	0.08	0.09
Bhp Hr/Gal	2870	3270	3040

■ Summary

The following points are of primary importance in the consideration of the problem of piston rings and oil control in two-cycle diesel engines:

(1) The piston skirt and compression rings should be lubricated sparingly to avoid excessive deposits.

(2) Oil must be controlled below the air intake ports.

(3) The piston-pin hole must be closed off to keep oil from flooding the piston skirt above the oil rings.

(4) The piston skirt and compression rings should carry their lubricant with them.

(5) The oil rings should be well vented and have unit pressures high enough to control a large amount of oil.

(6) Clean air flowing past the oil rings aids in controlling the oil, and in keeping the oil passages clear.

Requirements for CARBURETOR AIR FILTERS for Aircraft Engines

by WAYNE D. CANNON
Wright Aeronautical Corp.

AFTER having lived in a world of false security for many years, the aeronautical industry has finally begun to recognize its need for a satisfactory air filter in its engines. This awakening has been due largely to the present war, which is requiring that airplanes be operated from all kinds of fields in every part of the world. Desert warfare demands efficient carburetor air filters even for only a few hours of successful operation of an airplane.

In this paper Mr. Cannon discusses, with the help of illustrations, various makes of filters, including a crude but ingenious desert-made airscoop, a none-too-efficient German filter used on the Messerschmitt 109, and various installations designed in this country.

Our designs are quite different from the British and German ones, because we were able to take advantage of the extensive development work of the automobile and air conditioning industries.

One type of filter was found especially easy to adapt to aircraft. It was the viscous impingement filter, which is capable of very efficient operation.

The author has also included descriptions of a number of installations using the viscous impingement filter, which are believed to give the engine the protection it needs against damage by sand, with the least complication and the least added weight, without upsetting the carburetor metering, and without imposing any added duties on the pilot.

* * *

THE AUTHOR: WAYNE D. CANNON, graduate of New York University, 1931, has been employed at Wright Aeronautical Corp. for several years in the Field Engineering Division. His experience with engine operation and installation problems has taken him to South America, the Arctic and Europe. At present he is in charge of accessory installation problems for the Wright company.

UNTIL recently, there was probably a more general appreciation of the effect of sand and dirt on internal-combustion engines among farmers in this country than in the aeronautical industry. The farmer lived with the problem for years until the automotive industry tackled the job and developed a satisfactory filter to protect the tractor, truck, and passenger car engine from this form of damage.

The farmer could not forget the problem because sand not only affected his engines, but seeped into his house through closed doors and windows.

The aeronautical industry has been living in a sense of false security by virtue of the successful operation of the domestic airlines, which have experienced relatively little trouble attributable to sand and dust.

This success was possible because the airlines operated from well equipped bases, and not because the aircraft-engine designer had been able to produce an internal-combustion engine that could successfully digest sand.

Engine manufacturers have sensed the need for air filters more keenly than other branches of the industry; and as long ago as the Lindbergh Paris flight complete induction systems with filters were supplied as standard engine parts.

With the demand for higher power of engines and higher performance of aircraft, the engine manufacturer was asked to omit the portion of the induction system that was external to the engine. This was a natural development, since the carburetor airscoop and other external parts, while affecting the performance of the engine, also affected the performance of the airplane.

During this era of striving for maximum performance in aircraft, filters were left out of powerplant installations, and have not been replaced until recently, since customers' needs did not seem to require them.

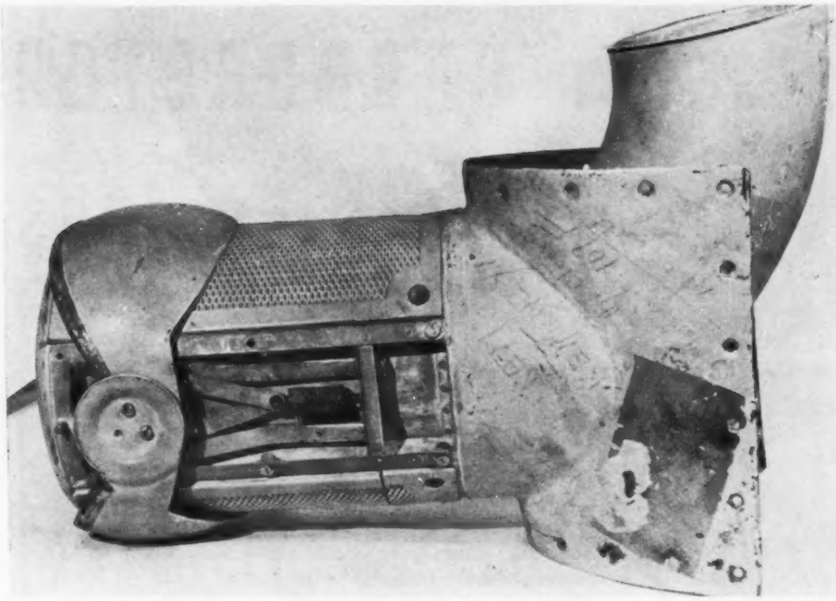
After many years of successful operation without carburetor air filters, it is not surprising that isolated cases of engine damage from sand and dust were overlooked when airplanes were on maneuvers.

Maneuvers, unlike wars, are soon over, or may be shifted to cover a wide variety of tactical problems, and thus some troubles never reach the malignant stage that requires a drastic remedy.

This is briefly the history of filter installations for aircraft engines in the United States.

There are, however, geographical areas in the world whose strategic value for the pursuance of military opera-

[This paper was presented at the National Aircraft Production Meeting of the SAE, Los Angeles, Calif., Oct. 1, 1942.]



■ Fig. 1 – Messerschmitt air filter and intake assembly

tions is not measured by the comforts the land can produce for its inhabitants. Some of these areas, usually deserts, are so important that a rigorous war must be carried on there in spite of the penalties that the locality imposes on men and machines. Out of these desert operations, there has been quickly relearned the vital importance of carburetor air filters; for without them aircraft engines can only be operated successfully for a few hours at the most, before piston-ring wear becomes a serious problem.

All combatants were faced with the necessity of providing a remedy, or else maintaining abnormally large maintenance and overhaul facilities with large quantities of spare equipment. Even if the spare-equipment situation were not a factor, which it is, the loss of aircraft due to sand damage to engines could not always be avoided. It is quite possible for piston-ring wear to progress so rapidly, with consequent high engine oil consumption, that the ship's oil supply would not be sufficient for a single flight, and the aircraft would have to be abandoned over enemy territory.

The Germans tried various types of filters; and it is reliably reported that none of their installations are very successful.

One of the special filter air intakes used on the Messerschmitt 109 has been received in this country. The filter had not been cleaned after its removal from the airplane, and it was therefore possible to evaluate how well the unit removes sand from the carburetor air.

To illustrate the configuration of the air intake assembly, and to better describe its use, Figs. 1 and 2 have been prepared.

In Fig. 1 it should be noted that the cylindrical intake assembly is faired into the leading edge of the wing and projects forward approximately 12¼ in., to expose the two curved filter elements which form the sides of the cylinder.

The end of the cylinder which projects directly forward from the wing is equipped with operable clamshell shutters, and at the discretion of the pilot these shutters may be closed to force air into the induction system through the filters. The control mechanism is visible in Fig. 1.

Fig. 2 shows the intake assembly with one of the filter elements removed, and sectioned to show its construction. The element consists of six layers of crimped dural, with the layers so disposed as to make a tortuous path for the entering air. The element is a dry type, and was undoubtedly designed with the idea that the inertia of the sand particles would carry the particles out of the air stream as the air turned into the filter.

Unfortunately, the photograph does not show the sand that was coated on all of the surfaces of the filter element, and that was deposited around the joints where the element is attached to the cylinder. Sand was found both inside and outside the intake assembly.

The sand appears to have entered the inside of the intake assembly during the time that the carburetor air was being taken in through the end of the cylinder, or in other words when the pilot moved the controls to bypass the filter.

It is probable that the principal reason for the presence of sand inside the Messerschmitt intake assembly is the inefficiency of the filtering media to extract from the carburetor air the sand that is actually present. There are, however, two other factors that may have contributed to

■ Fig. 2 – Messerschmitt air filter partially disassembled

this condition. The flying duties of the pilot may have so absorbed his attention that he forgot to place the filter in the air circuit before landing. It is also possible that the sand entered the intake assembly in flight, since the human eye cannot be relied on always to detect the presence of damaging concentrations of sand in the air.

At this point it might be well to consider the nature of the material that can enter the aircraft engine with the carburetor air and cause mechanical damage. It is equally important to consider where this material may be encountered during the operation of the airplane.

Everyone is aware of the dust that is prevalent on some landing fields and which is sometimes blown into the faces of spectators as a ship is maneuvered away from a station. It is obvious that some dust gets into the airscoop during these ground operations, particularly if several airplanes are operating in unison from the field.

Many people have noted the copper-colored clouds in the dust bowl area of the United States produced by dust laden air; and it is common knowledge that lava ash from volcanic eruptions ascends to high altitudes and remains suspended for long periods.

Although sand covers large areas of the world's surface, most people have a false impression of its composition.

The foreign material that must be removed from carburetor air by the aircraft-engine filter may vary considerably, but sand is probably the most destructive of these agents.

The chemical analysis of a typical desert sand showed that silica was the principal constituent, and that small percentages of calcium, magnesium, iron, and aluminum were present.

The sample was then analyzed for grain size and structure, with the following results:

Forty-seven per cent of the sample washed out in a standard clay test.

The remainder of the sample was screened. The percentages noted refer to the percentage by weight of the sample which was retained on each screen as the mesh was progressively reduced from No. 30 to No. 325.

U. S. standard screen No. 30 –	0.04%
40 –	0.04
50 –	0.10
70 –	0.10
100 –	0.40
140 –	1.10
200 –	4.24
270 –	8.24
325 –	37.80

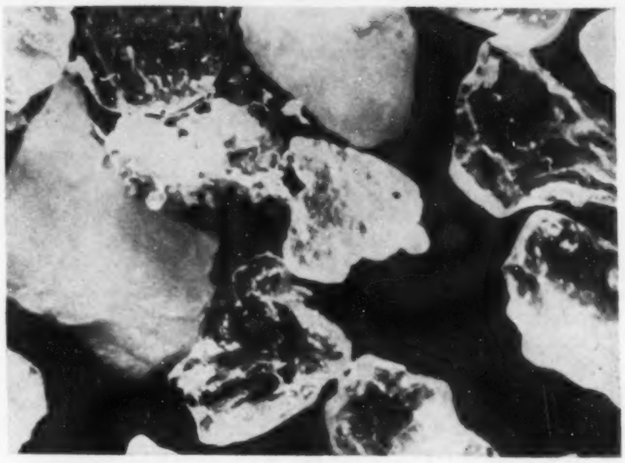
The portion of the sample washed out in a standard clay test was further analyzed under a microscope and found to be made up of particles, each having the same angular and subangular crystalline structure as the larger grains. The smallest particle measured 20 microns, which is approximately 0.00078 in.

To better illustrate the composition of the sand, Figs. 3 to 6 have been prepared showing the original sample and the Nos. 40, 70, and 270 screenings.

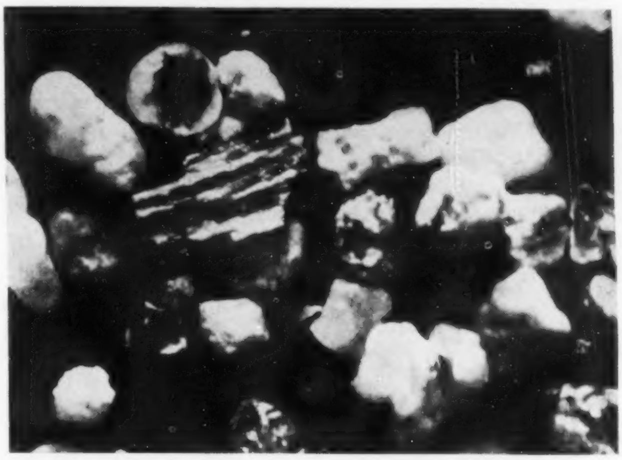
Two additional tests were made to evaluate the abrasiveness of the sand. The first test consisted of placing some of the sample in a hydraulic press; and it was found that the particles fractured along cleavage lines to produce smaller but more jagged crystals.

The second test was made by rubbing a piece of cast iron on a nitralloy block of 900 Brinell hardness which

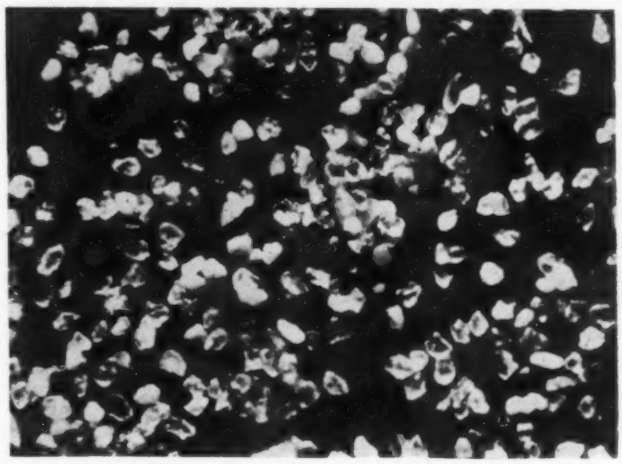
■ Fig. 3 – Typical desert sand sample



■ Fig. 4 – Analysis of sand sample – No. 40 screening



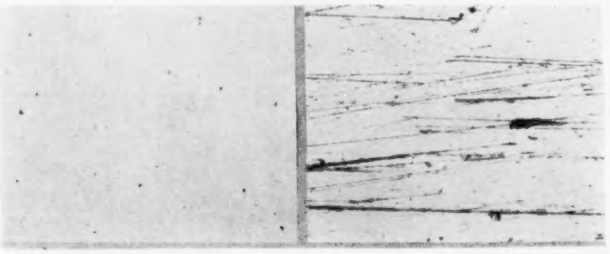
■ Fig. 5 – Analysis of sand sample – No. 70 screening



■ Fig. 6 – Analysis of sand sample – No. 270 screening

had been coated with a small portion of the original sand sample mixed with lubricating oil.

After a few strokes made with finger pressure, the nitralloy was scratched as shown in Fig. 7, which was made under 100 magnifications.



■ Fig. 7 – Abrasion test. View of polished nitralloy block – before and after test (reduced from photomicrographs taken at 100X)

Having shown by analysis the destructive qualities of this sand, it is of equal importance to know that flight tests have verified its presence at 12,000 to 15,000 ft.

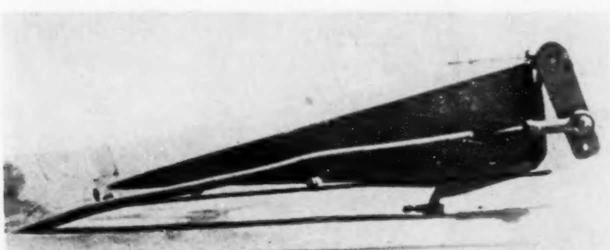
There have been many filter designs evolved and tested to protect engines from sand damage. One of the German designs has been discussed. The British designs have been well established for many years. The Vokes Company of England has been the principal manufacturer of filters preferred by the British. Its design employs the use of gauze supported between layers of wire mesh formed into pleats, in accordion fashion. The pleating is provided to increase the area of the filter that will be exposed to the air in a scoop of a given cross-section. This increase in area is important since the filter is installed in the main air intake of most British installations without bypassing provisions.

This type of filter has the advantage of being lightweight and easy to handle. It can be made as a dry filter, or it can be dipped in a special low viscosity oil for use as a wet element.

The dry type of filter of Vokes design was adapted to a number of American-made aircraft used by the British. Some of the early installations were made in the desert.

The desert is not the place to redesign or rebuild airplanes. Materials are scarce, and fabrication facilities are scarcer. What is more important, time is not available for untrained personnel to experiment with airplane designs.

It is difficult to visualize the extreme measures that were taken when filters were built out of any material available and installed with a minimum of delay. Fig. 8 is a photograph of such an improvised filter installation which is a credit to the men who built it.



■ Fig. 8 – Desert-made airscoop

After showing Fig. 8 it hardly seems necessary to explain why the filter installation problem rests primarily with the aircraft manufacturer, and even though the photograph bears no resemblance to equipment now in use, this desert-made airscoop should remain as a challenge to the ingenuity and skill of American aircraft designers to continue to produce efficient filter installations in high-performance airplanes.

The filter problem has been approached in this country in a way that is different from either the German or the British method.

This approach was possible because extensive work had been successfully done by the automobile and air conditioning industries. It was a relatively easy matter to choose proven filter media and adapt them to aircraft use.

The complete story of how many promising designs were selected and tried probably will never be written, although it is very interesting. At this time it is enough to say that some designs were discarded because of weight, or because they were not sufficiently rugged to meet American aircraft standards; others because they imposed too great a restriction to airflow in a high-velocity duct, or lost their efficiency under these same conditions; and others were discarded because of undue complication to the induction system.

Viscous impingement filter elements, however, were found to be easily adaptable to aircraft. These filter elements have been described in detail in several publications. Briefly, they consist of knit wire or knit metal ribbon sheets suitably crimped and packed into durable metal frames. The mesh of the sheets is varied, becoming progressively smaller from the entering face to the rear face of the element. The layers are supported in the frame with sufficient structure to withstand engine backfire pressures without damage.

Before use, the filter is dipped in engine lubricating oil and allowed to drain before it is installed in the induction system. This oil entraps and holds the sand after the filtering media mechanically separate the particles from the air stream. The filter is easily serviced by cleaning it in gasoline and redipping it in oil.

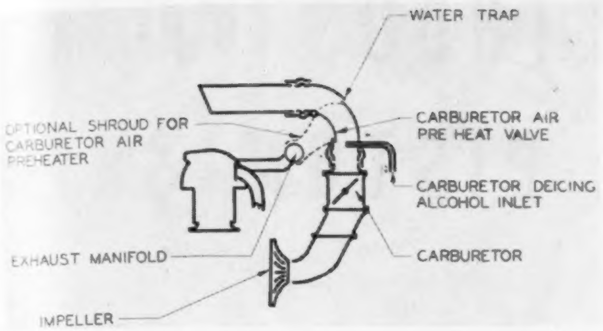
Viscous impingement filters when properly made are capable of removing a minimum of 90% of the sand from the air when the air velocity is 1000 fpm without imposing a pressure drop of more than 4 in. of water.

Filters having this performance can be built in a wide variety of shapes to fit the installation requirements, although flat surfaces are desirable for manufacturing reasons. The thickness of the element has been standardized at 2 in.

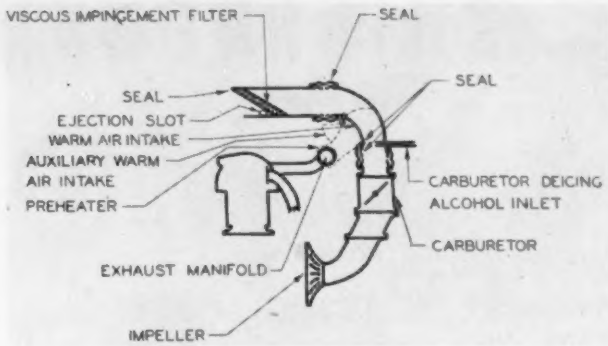
No attempt has been made to standardize the dimensions of a filter element because the configuration of airscoops varies with the design of the airplane, and therefore the shape of the filter has been left flexible so that installation designers will have as much leeway as possible.

Even though the design treatment of an induction system varies widely from one airplane to another, there are certain parts which are common to them all which are shown diagrammatically on Fig. 9.

The essential parts shown are the main air intake, the carburetor air preheat valve, the carburetor de-icing alcohol inlet or the optional exhaust manifold shroud for a carburetor air preheater, and a water trap which is incorporated in many scoops. The rest of the induction system is a part of the engine.



■ **Fig. 9 – Installation diagram of a typical non-filtering induction system**



■ **Fig. 10 – Installation diagram of an induction system with a viscous impingement filter installed in the main air intake (exhaust manifold)**

There are several ways in which the conventional induction system can be altered to accommodate a filter element. Several methods have already been discussed. Fig. 10, however, illustrates diagrammatically how the conventional induction system should be changed to install an air filter properly.

Note first that the filter is disposed so that it can be easily removed for cleaning. Ease of servicing is of great importance.

It is usually necessary to position the filter at an angle to get an element of sufficient area into the scoop. The drop through the filter increases inversely as the angle of incidence decreases, but the drop is not too important above 20 deg.

Just ahead of the filter element are ejection slots which bleed off some air which carries away the dirt and free water that settles in the airscoop or is deflected by the filter element. These slots should be located with respect to pressures inside and outside the scoop so that the airflow will always be in the proper direction.

These slots replace the water trap shown on the previous system, since it is of utmost importance that the induction system be thoroughly sealed to prevent sand bypassing the filter, as would be the case if the water trap were not omitted.

In this connection it can be pointed out that seals are shown around the filter element, at all joints, and at the alternate air intake.

The rest of the system is the same as a non-filtering installation with one exception. That is the treatment of the alternate air intake and carburetor air preheater.

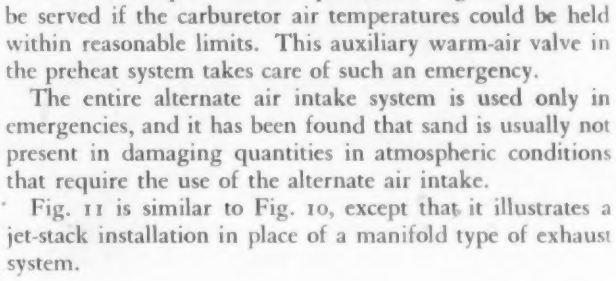
In Fig. 10, the solid lines represent the alternate air intake which takes warm air from the rear of the engine. The dotted lines indicate a shrouded exhaust manifold which picks up and preheats the carburetor air before it enters the induction system. This system may produce very high carburetor air temperatures if there is not an admixture of cold air through the main air intake. For this reason another valve has been added, which will be referred to as the auxiliary warm-air valve. This valve can be rigged to a single preheat control in the cockpit, so that for the first portion of the cockpit control travel the auxiliary warm-air valve remains fully open, and does not start to close until the admission valve in the airscoop is fully open. As the cockpit control is moved further toward the full-hot position, the auxiliary warm-air valve closes to give the pilot full use of the preheater.

This method of operating the preheat system adds no duties to the pilot and yet gives him an induction system suitable for flying in all kinds of weather.

This additional warm-air valve may appear to be an unnecessary refinement. It has been used on commercial airplanes for many years and is only new in its application to a filter installation. It is important, however, since it is conceivable that a ship with a filter installed in the airscoop may climb through a snow storm and have the filter completely blanked off and come out into sub-freezing temperatures that will not clear the filter. If, under these conditions, full power was required, the engine could best be served if the carburetor air temperatures could be held within reasonable limits. This auxiliary warm-air valve in the preheat system takes care of such an emergency.

The entire alternate air intake system is used only in emergencies, and it has been found that sand is usually not present in damaging quantities in atmospheric conditions that require the use of the alternate air intake.

Fig. 11 is similar to Fig. 10, except that it illustrates a jet-stack installation in place of a manifold type of exhaust system.



■ **Fig. 11 – Installation diagram of an induction system with a viscous impingement filter installed in the main air intake (jet stacks)**

Fig. 12 illustrates a filter installation in an induction system employing a turbosupercharger.

Note that the filter is located on the pressure side of the

turn to page 63

NEW METHODS for the E
of PISTON-SKIRT D

ALTHOUGH a lubricating oil or an additive may originate in the chemical laboratory and be subjected to many physical and chemical laboratory tests, the final proof of its suitability comes as a result of engine tests under conditions simulating those that the oil must meet in service. These tests, to be of value, must be carefully controlled, and the results accurately evaluated and recorded in a form that is reproducible.

This task is especialy difficult when piston-skirt deposits are to be considered. The bare statement that pistòn *A* has more deposits than piston *B* is not sufficient. Quantitative results must be available: first, when a given engine test is developed, so as to determine the degree of reproducibility attained by the test; and second, when comparisons are to be made between different lubricants. Added advantages of quantitative measurements are, that they make possible comparisons between separate laboratories and facilitate the standardization of test procedures.

Specifications for such measurements should include:

1. Numerical evaluation of piston-skirt deposits that could be reproduced by different observers even though not skilled in the procedure used.

2. A method of evaluation that uses commercially available equipment, in order that any laboratory could adopt the procedure with minimum trouble.

3. An objective system of measurement, so that the method will not be subject to major revision by each individual observer.

In addition to the problem of measuring the deposit on the piston, there is the problem of photographing the form, location, and intensity of these deposits, in order to keep a record available without the need for preservation of all used pistons. Photography of a cylindrical object is not easy if all direct light reflection from the surface is to be avoided; and photography of a piston surface to show the entire circumference on one print with uniform lighting that accentuates the carbon deposits and does not distort portions of the surface is desirable.

It is the purpose of this paper to describe two devices now in use in the authors' laboratory that greatly assist in the solution of these problems.

THE final proof of whether or not a lubricating oil is suitable for use, comes only as the result of engine tests conducted under conditions simulating those the oil must meet in service. To be of value, these tests must be carefully controlled and the results recorded in a form that is reproducible.

In the case of piston-skirt deposits, it has been found extremely difficult to secure accurate quantitative results, reproducible by different observers; and to obtain a permanent record of the form, location, and intensity of the deposits.

The authors' work on piston-skirt deposits has led them to develop two devices, which they have described here in detail.

The first of these makes use of a reflection densitometer to obtain readings of the reflection density of a large number of points on the piston to be studied. These readings are converted to

■ ■ ■

THE AUTHORS: H. R. LUCK, head photographer for the Shell Development Co., Emeryville, Calif., is a graduate of the University of California, and has been active in various phases of photography for the past 14 years. Before joining the Shell Co. three years ago, he was color expert for a Pacific Coast gravure concern. His special field is color reproduction, and as a result of his experience in color work, the piston photography described in the paper has been extended to include "direct separation" color reproduction of pistons. T. A. ROGERS was graduated from the University of California in 1928 as an electrical engineer and returned to teach electrical courses while studying for a Ph.D. He joined the Shell Oil Motor Laboratory staff at Martinez, Calif., in 1938, and later transferred to the Shell Development laboratories in

■ Evaluation of Piston Lacquer Deposits

Piston lacquer deposits have been evaluated by direct comparison with standard pistons of graded density[1] as a means of assigning a value to the deposit. This is a very convenient and rapid method but it has the disadvantages of depending upon the judgment of the observer, and of fairly large steps between successive standards. If an attempt is made to reduce the deviation between successive engine tests or to compare two lubricants of slightly different characteristics, considerable difficulty may be met in describing exactly the piston deposits.

Of great assistance was found to be the application of

[This paper was presented at the National Fuels and Lubricants Meeting of the SAE, at Tulsa, Okla., Oct. 23, 1942.]

[1] See *National Petroleum News*, Vol. 32, No. 36, Sept. 4, 1940, pp. R-314-R-320: "Show Motor Oil Varnish Tendencies with Single-Cylinder Test Engines."

EVALUATION and RECORDING DEPOSITS

by H. R. LUCK, T. A. ROGERS, and A. G. CATTANEO

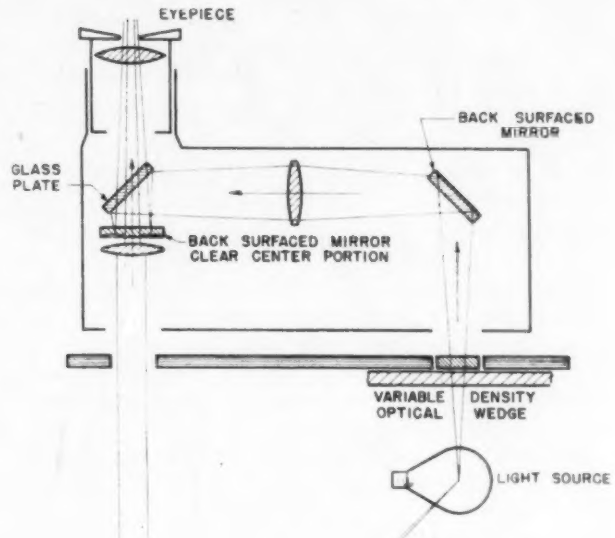
Shell Development Co.

reciprocal reflectivity values, the average of these values being called the piston rating. A year's experience with this method has shown that the results are reproducible by different observers and are most helpful in evaluating the piston deposits.

The other device described is a piston photographing machine, which attempts to overcome the difficulties inherent in the photography of cylindrical objects. The apparatus is based on the principle that the piston, when rolling in a circle, at some time presents each point of its circumference equidistant from the center of the circle. The instrument gives a photograph of the entire piston surface developed upon a plane with practically no distortion. Even photographs in color can be obtained, thus allowing direct comparison not only of amount and type of deposit but also of the color. Simplicity of operation and uniformity of the conditions for piston lighting and exposure are the main advantages of this machine.

■ ■ ■

Emeryville, where he is now conducting research on engine lubrication. He has published several papers in the electrical engineering field, two of which combine engineering and his hobby of photography. A. G. CATTANEO (M '38) was graduated from the Federal Technical University of Zurich, Switzerland, and later received his degree of Doctor of Science from the Karlsruhe University, Germany. From 1934 to 1937 he was a member of the Delft Laboratory of the Royal Dutch Shell Group where he did research on detonation and other engine fuel problems. He joined the staff of the Shell Development Co. at Emeryville in 1937 and organized an engine research laboratory which is now full grown and carries on investigations on all phases of fuel and lubricant development.



■ Fig. 1 – Schematic diagram of the Eastman Reflection Densitometer

an Eastman Reflection and Transmission Densitometer[2, 3] which was developed by the Eastman Kodak Company for use in the graphic arts industry. The principle of the instrument is to match the intensity of two light beams. Fig. 1 shows schematically the common source of light in the instrument, one part of which is directed to a small area of the object to be tested and another part through a calibrated optical absorption wedge. The two pencils of light are brought together at a mirror surface to allow direct comparison of their intensities. Comparisons are made by matching the intensity of the light passing through the calibrated wedge against the light reflected from the piston surface at the spot under consideration. The light

[2] See *Journal of the Optical Society of America*, Vol. 24, No. 1, Jan., 1934, pp. 19-24: "A Reflection Densitometer," by J. W. McFarlane.
[3] See *Journal of the Optical Society of America*, Vol. 25, No. 12, Dec., 1935, pp. 417-419: "The Eastman Transmission and Reflection Densitometer," by C. A. Morrison and J. W. McFarlane.

passing through the wedge can be varied by moving the wedge from a position of zero absorption to a position of almost 100% absorption, or until the intensity matches that reflected from the surface of the object.

In the evaluation of piston deposits, initial calibration is obtained by adjusting the light source for balance on a standard white card (Eastman Color Temperature Test Paper) with a red viewing filter over the eyepiece. The red filter reduces the difficulty of matching the intensity of the gray light from the optical wedge with the reddish-brown light reflected from the piston-skirt surface. The red filter may be thought to discriminate against the very light brown lacquers by giving a lower rating than without a filter. However, the use of a light blue filter for contrast to accentuate the reds or browns does not increase the ratings appreciably for the light browns. Therefore, the effect of the red filter is negligible over that region where one might expect the greatest change. Of course there is an appreciable increase for the darker red-brown deposits.

A numerical scale attached to the optical wedge is calibrated in terms of reflection density.

The results, as outlined in Appendix I, are expressed in terms of the reciprocal reflectivity, $1/R$, where $R =$ the reflectivity of a surface under 45-deg illumination. Reciprocal reflectivity is related[4] to reflection density by the expression:

$$1/R = 10^D = \text{antilog } D$$

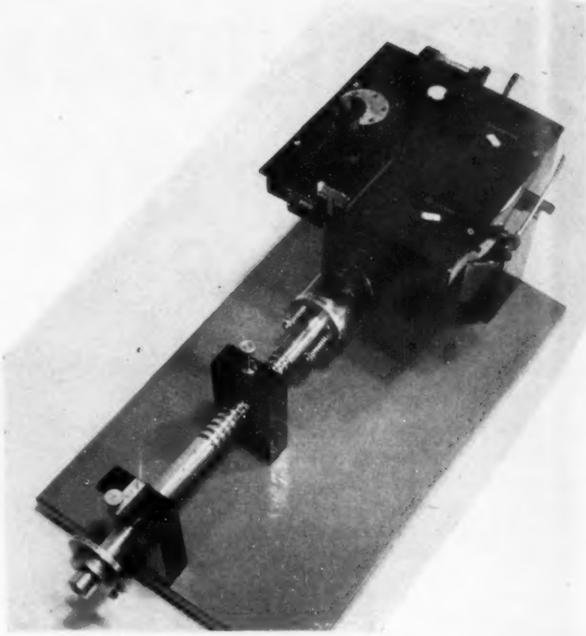
If the surface is perfectly white and is a perfect diffuse reflector, then $R = 1$ and $D = 0$; and if the object is quite black, then $R = 1/100$ and $D = 2$. The range from $D = 0$ to $D = 2$ which is covered by the densitometer includes, for practical purposes, the entire density scale of piston deposits.

The piston to be evaluated is scanned at a predetermined number of points and the readings converted to reciprocal reflectivity values. The average of these values is called the piston rating. A clean piston may have a rating very near 0% and an extremely black piston may approach 100%.

The final rating is the numerical difference between the rating of the clean piston as it goes into the engine and the lacquered piston as it comes out of the engine at the end of a test.

A densitometer installation is illustrated in Fig. 2 with a small test-engine piston in place for rating. Proper indexing is obtained by means of the slotted and drilled rod holding the piston. A series of 40 readings is taken. An example is shown in Table 1. The piston rating computed from the values shown is 2.74.

[4] See "The Principles of Optics," by A. C. Hardy and F. H. Perrin, McGraw-Hill, New York, 1932.



■ Fig. 2 – Densitometer installation for rating lacquer deposits on a small piston

Piston ratings have been made with the densitometer for about one year with highly satisfactory results. It is possible for any one of the engineers in charge of the tests or any of the engine operators conducting the work to obtain almost identical values from the same piston. In a typical case, measurements were made by four different persons on the same piston with the resultant ratings of 2.74, 2.82, 2.78, 2.72. It is felt that the problem of reproducibility in piston rating is fairly well solved by this method.

The advantages of such a method are readily appreciated by those undertaking engine tests on lubricating oils, and it is in order, therefore, to discuss some of the disadvantages or difficulties that are apt to be encountered.

Table 1 – Densitometer Rating of Piston Lacquer

Piston Positions Length Circumference, Deg. from Thrust Side	1		2		3		4		5	
	Density	1/R	Density	1/R	Density	1/R	Density	1/R	Density	1/R
0	0.83	6.76	0.54	3.41	0.23	1.70	0.21	1.62	0.43	2.69
45	0.75	5.62	0.91	8.12	0.48	3.02	0.40	2.51	0.56	3.63
90	0.70	5.01	0.90	7.94	0.53	3.38	0.33	2.14	0.46	2.88
135	0.74	5.49	0.66	4.57	0.39	2.45	0.42	2.63	0.54	3.46
180	0.78	6.02	0.65	4.47	0.30	1.99	0.33	2.14	0.35	2.24
225	0.87	7.41	0.65	4.47	0.54	3.46	0.30	1.99	0.40	2.51
270	0.66	4.57	0.70	5.01	0.40	2.51	0.21	1.62	0.34	2.18
315	0.82	6.61	0.74	5.49	0.36	2.29	0.17	1.48	0.28	1.91

Average Value of 1/R = 3.74
Average Value of 1/R for unused piston = 1.00
Average Gain in 1/R = 2.74 Piston Rating
R = Reflectivity

The first difficulty will occur if the deposits on the piston follow a definite pattern which coincides with the densitometer observations on the piston. In such a case, additional observations in different positions should be made. Likewise, if all the points selected for measurement fall on areas of thin deposits or on areas of extreme deposit, an erroneous reading will be obtained. Such errors can be overcome by selecting a greater number of points.

A second difficulty enters when a part of the area to be rated is lightly scratched or scuffed. The rating for an area that is scuffed, even though free from deposits, may indicate that the area is darker than the clean piston. This is due to uneven scattering of the light from the scuffed surface. All readings from scuffed areas should be disregarded, and the average determined from consideration of the remaining values. In fact, if the major portion of the piston surface is scuffed, it will be impossible to obtain a correct densitometer rating. This is not of great importance, since a lubricant which allows scuffing to this extent is, for this reason alone, not satisfactory.

A third difficulty, for which there appears to be no immediate remedy, occurs when an area of the piston is coated with a transparent lacquer. This condition is found occasionally in work on highly refined lubricating oils and can only be detected by close surface examination of the piston. Such pistons read as though they were clean.

In spite of these difficulties, the method has been found so useful that all of the important test-engine pistons in our laboratory are being rated by this means.

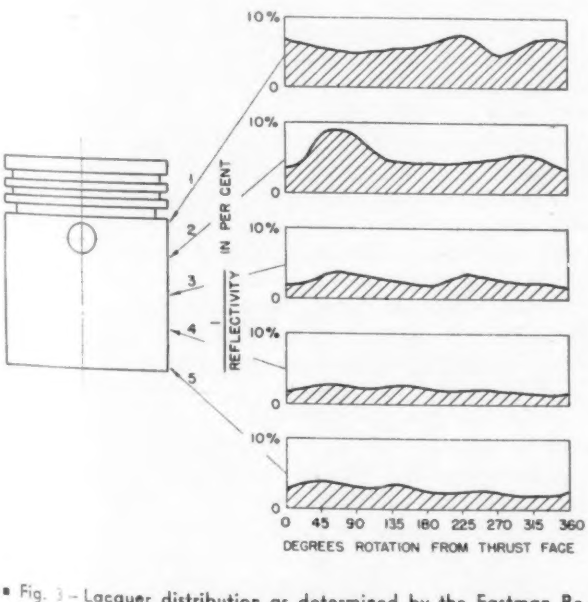
The most practical use of individual point ratings has been found to be that of averaging the values for all the points to obtain a final average value. Another possible means of representing the piston deposit is by graphing the density values as a function of location on the skirt such as shown in Fig. 3, but this is time-consuming and is not used in practice.

An important control factor was brought to light by the use of the densitometer. Normal cleaning procedure left the piston in a successively poorer surface condition as testing progressed. Scuffed or polished areas were not removed and the piston was judged on the basis of overall discoloration. When the pistons were checked on the densitometer for initial reading, it was found that nearly identical surfaces could be obtained by polishing the skirt with a hand soap containing a mild abrasive. This action produced a uniform matte surface and gave an initial reading of practically zero density.

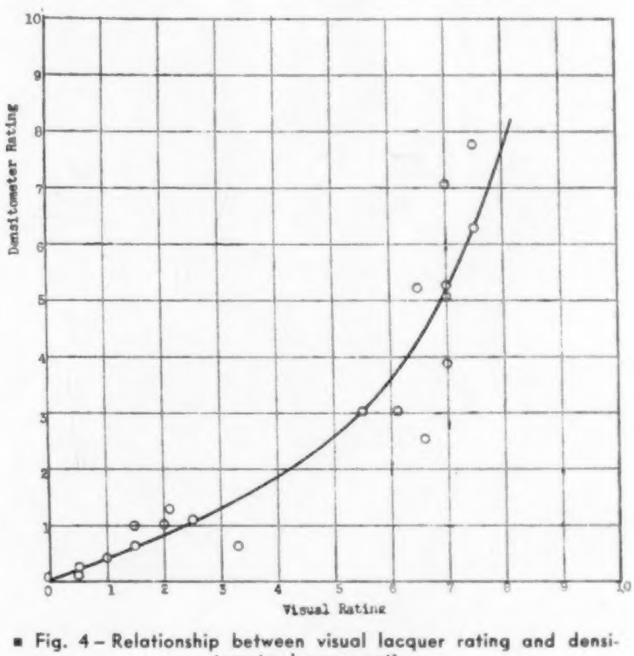
Another point to be considered is the method of removing excess lubricating oil from the piston surface before taking the densitometer readings. It has been found that some lubricant lacquer binders are soluble in straight-run gasoline and the deposit flakes off after cleaning and drying. This laboratory has followed two procedures, both of which are satisfactory. The first, used for small pistons, is to allow the piston to drain for about 24 hr before the rating is made. The second method, used in those cases where a heavy aviation grade oil is tested, is to dip the piston in a very light white oil until the aviation oil is cut to a low enough viscosity to drain off. The use of a leaded gasoline produces a white deposit over the piston surface that results in too light ratings when the piston is cleaned in clear gasoline, but no deposit is visible, nor are ratings affected, when the piston is cleaned in white oil of USP grade.

Finally, it should be pointed out that the densitometer rating is subject to the criticism that it evaluates the surface appearance rather than the thickness of the lacquer. Some few measurements on lacquered Lauson engine pistons, whose lacquer was carefuly rubbed off and weighed, have shown good agreement with the densitometer rating.

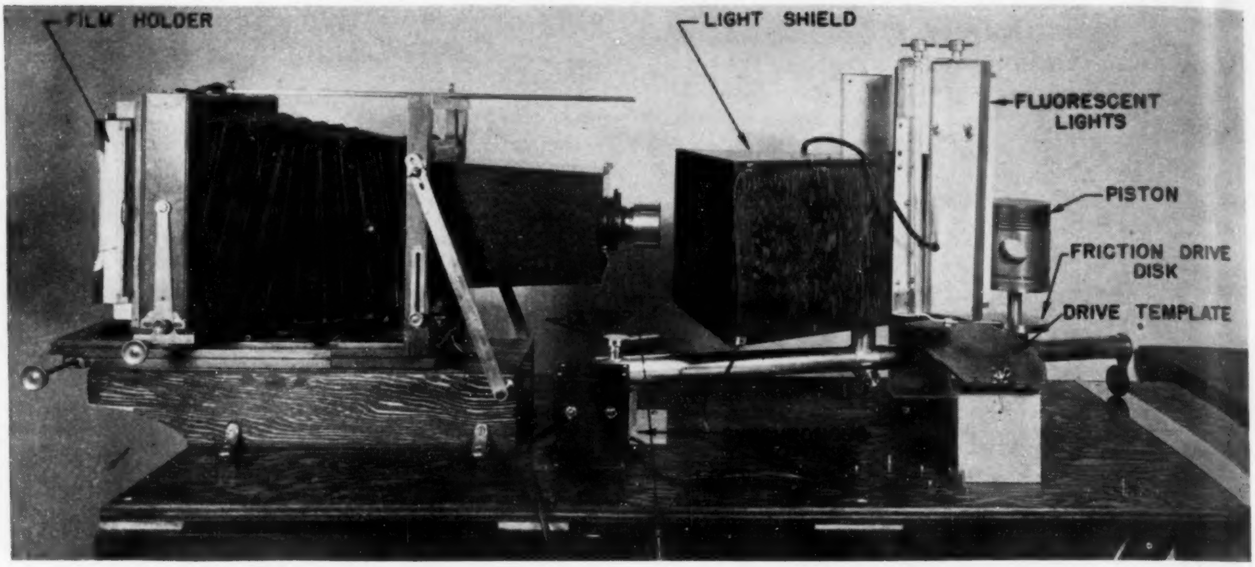
Between densitometer ratings and visual ratings an average correlation curve can be established. Fig. 4 shows this correlation for a visual method in which the shades of the lacquer, rated 0 to 10, are multiplied by the percentage of area covered, giving an average lacquer shade. Deviations from this curve are nearly always due to the



▪ Fig. 3 – Lacquer distribution as determined by the Eastman Reflection Densitometer



▪ Fig. 4 – Relationship between visual lacquer rating and densitometer lacquer rating



■ Fig. 5 – Piston surface camera and attachments

individual observer's difficulty with the visual method.

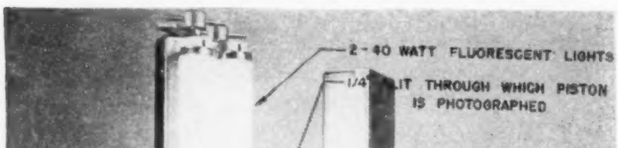
On the basis of our experience with this instrument, we would not at this stage recommend it for general adaption for routine work, since the rating obtained must be carefully interpreted in the light of the particular characteristics of the surface under examination. However, for the research laboratory designing a reproducible method for lacquer rating, the instrument has possibilities well worth consideration.

■ Piston Photography

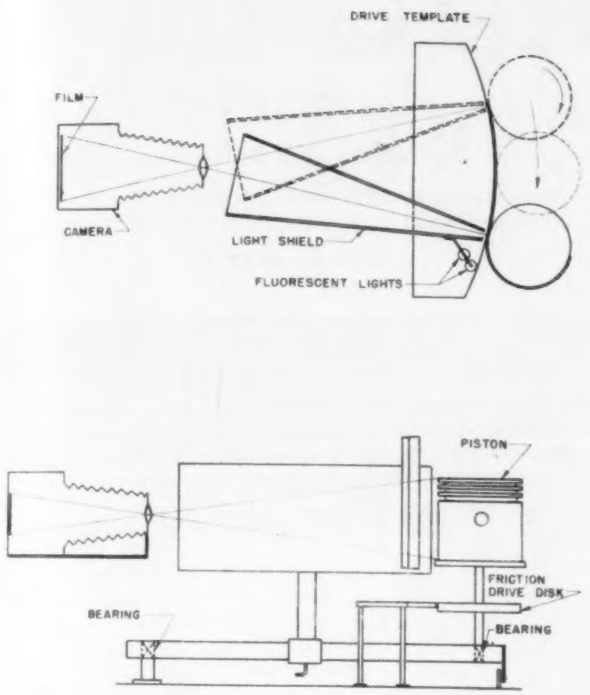
Photographs of pistons and engine parts are very useful in recording the overall appearance of the subject matter. Too often, however, a true representation of the surface deposits is difficult to obtain, and a correct representation of a curved surface by conventional procedures is next to impossible. All attempts to photograph a cylindrical surface have introduced errors of distortion or errors in lighting that do not truly represent the density of the piston deposits. Frequently, a light source reflecting into the camera lens from a smooth jet-black surface will produce the appearance of clean metal instead of a black lacquer deposit.

In an attempt to eliminate these difficulties, a piston photographing machine was developed that gave a final image of the entire piston surface developed upon a plane with practically no distortion. This device is based upon the principle that the piston, when rolling on a circle, at some time presents each point of its circumference equidistant from the center of the circle. This principle is embodied in a mechanism (Figs. 5 and 6) that photographs only the portion of the piston in contact with the circle and takes a continuous exposure while the piston is rolled along at a constant speed. Fig. 7 illustrates schematically the construction of the device.

As the piston rolls along the generating arc, only that portion of its surface is exposed which is within ⅛ in. on either side of the point of contact. This is accomplished by photographing through a slot ¼ in. wide which moves with the piston and is in line with the point of contact and the camera lens. A slit ¼ in. wide does not introduce an appreciable error in the image sharpness due to the motion of a point on the piston surface in approaching the point of contact and in leaving it. Other points off the surface, such as the bottom of the ring grooves, never become stationary during the exposure and hence have a blurred image. The speed of motion of the piston along the generating circle is adjusted in the initial design to provide a



■ Fig. 6 – Front view showing friction drive and lighting



▪ Fig. 7 – Diagrammatic sketches of the piston surface camera and attachments

normal exposure for the light source, lens opening, and film used. In the apparatus illustrated, the speed was such as to allow for a 5-sec exposure of any point. The action is similar to that of a focal-plane shutter at the plane of the object instead of at the plane of the image as in the conventional camera.

The function of this apparatus has been accomplished by other types of cameras[5, 6] constructed for a specific purpose. This device was designed to have as great a range of application as possible, and it accommodates pistons from 2⅜-in. diameter to 6⅛-in. diameter. It may be adjusted for pistons of other diameters as the need arises.

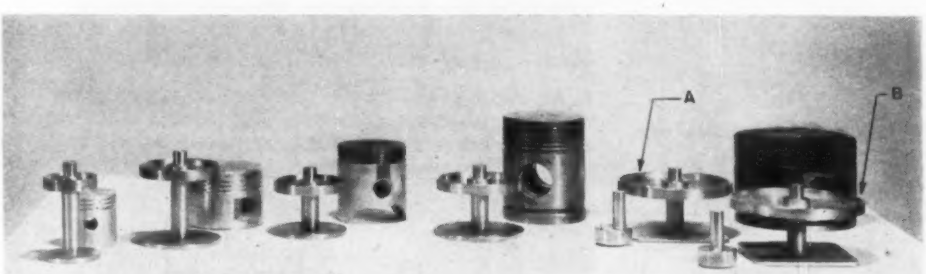
Fig. 8 illustrates the present series of pistons that the machine will accommodate, with their accompanying turntables. The turntable must have the same rolling diameter as the surface to be photographed, which in most cases is the piston skirt. However, as one of the important items of study on the large aircraft piston is the amount and type of deposit in the ring grooves, a second turntable is provided which has a diameter equal to that at the bottom of the ring grooves. When the piston skirt is to be photographed, turntable *B* is used; when the ring grooves are required, turntable *A* is used and is offset on the driving arm by the amount of the decrease in the turntable radius so as to use the same generating circle.

The type of photograph obtained by this device is somewhat startling at first, in that it allows one to see all sides of the piston at once, as shown in Fig. 9. Once the novelty has worn off, it is more convenient to photograph only half of the piston at a time (Fig. 10) as this gives a better image for comparison purposes. The conventional type of photograph is shown in Fig. 11. The piston illustrated in these three figures is from the Chevrolet test on the SAE correlation oil B-4. The test lasted only 29 of the 36 hr, as the piston-skirt varnish set during the shutdown period so the engine could not be started. The piston had to be driven out of the cylinder when the engine was taken down, leaving the lacquer deposit badly scuffed. The motion after each blow is shown clearly on the photograph.

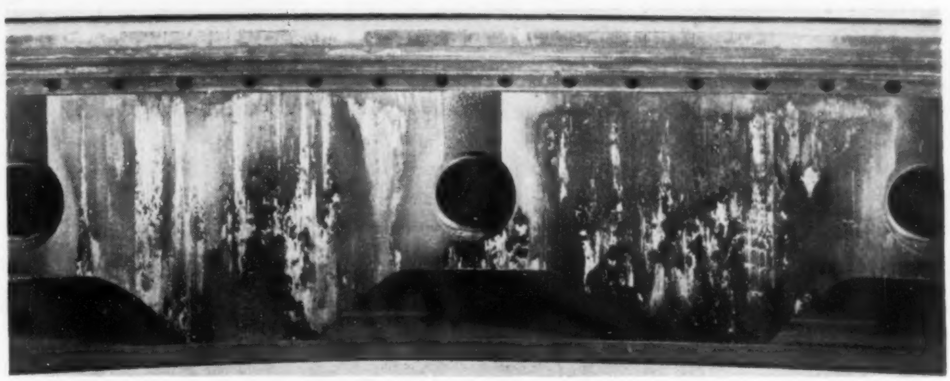
Figs. 12 and 13 show two views of a single-cylinder Caterpillar piston after operation on an unsatisfactory lubricant. The original finish marks, as well as all scuffed marks or scratches, are clearly visible. These two views

[5] See *Scientific Papers*, No. 517, "A Special Camera for Photographing Cylindrical Surfaces," by Raymond Davis, Bureau of Standards, Washington, D. C.

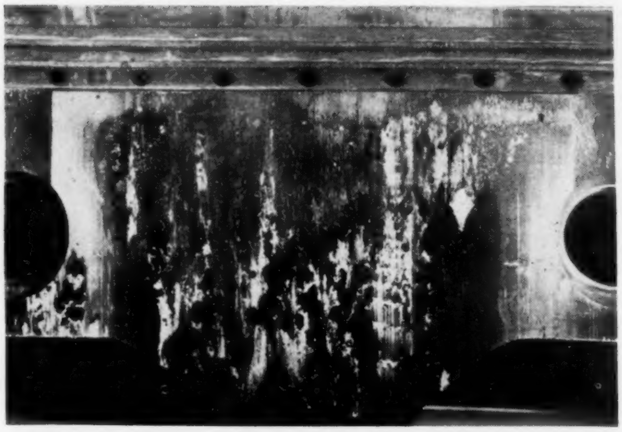
[6] See *Photography Handbook*, No. 2, 1938, p. 102: "Flat Pictures of Round Objects," Fawcett Publications, Inc., Greenwich, Conn.



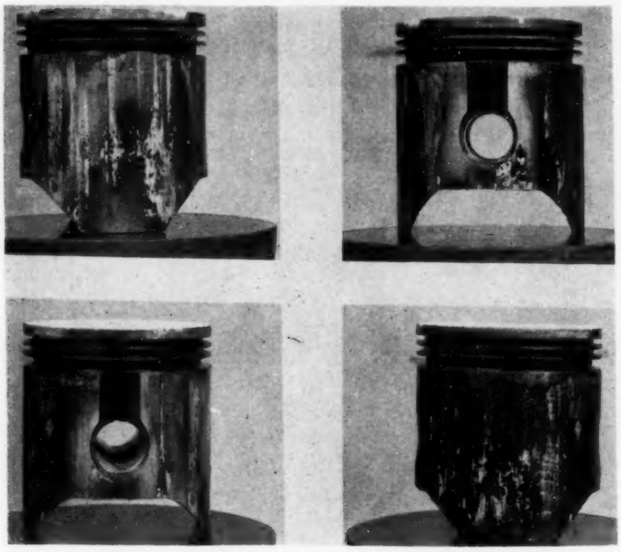
▪ Fig. 8 – Present series of pistons and their accompanying turntables



▪ Fig. 9 – 360-deg photograph of a Chevrolet piston after 29 hr on SAE reference lubricant B-4



■ Fig. 10 – Chevrolet piston run 29 hr of the standard 36-hr test on SAE reference lubricant B-4



■ Fig. 11 – Conventional 360-deg photograph of a Chevrolet piston after 29 hr on SAE reference lubricant B-4

show the carbon deposits on the lands and above the top ring in considerable detail.

With the proper light sources and filters, it is possible to obtain the same type of picture in color on Kodachrome film. This is a great advantage, as it allows direct comparison not only of amount and type of deposit but also of the color. Some of the photographic data are tabulated in Appendix II for the convenience of those investigators who may wish to construct a similar machine.

This particular design is not, of course, the only possible arrangement, but it has the advantage of simplicity, particularly from the standpoint of piston illumination. An alternative design would provide that the piston roll on a straight line perpendicular to the lens axis and thus keep the image in the same plane throughout the exposure. This has many advantages by way of a rectangular image, uniform depth of focus, and adaptability to many piston sizes. The main disadvantage lies in non-uniform illumination due to the varying distance of the object from the lens.

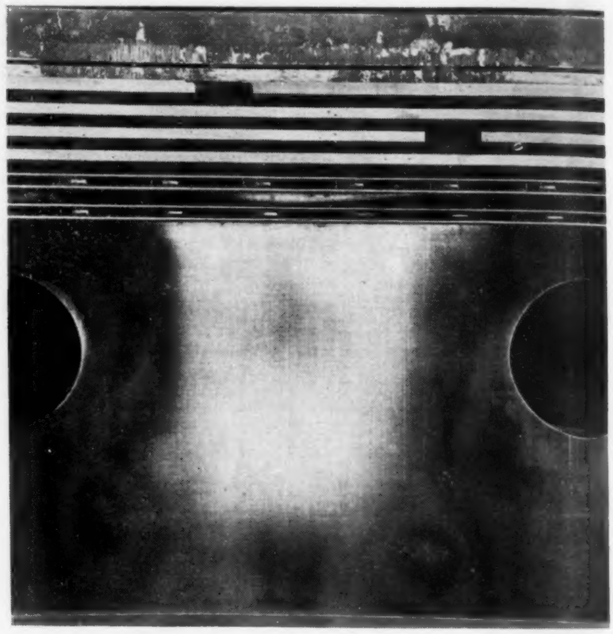
Probably the main advantage of the apparatus lies in the uniform conditions for piston lighting and exposure. It is a very simple matter to drop a piston in place, insert a plateholder and start the motor. The machine does the rest, and produces results that may be used for direct comparison either in black and white or in color.

APPENDIX I

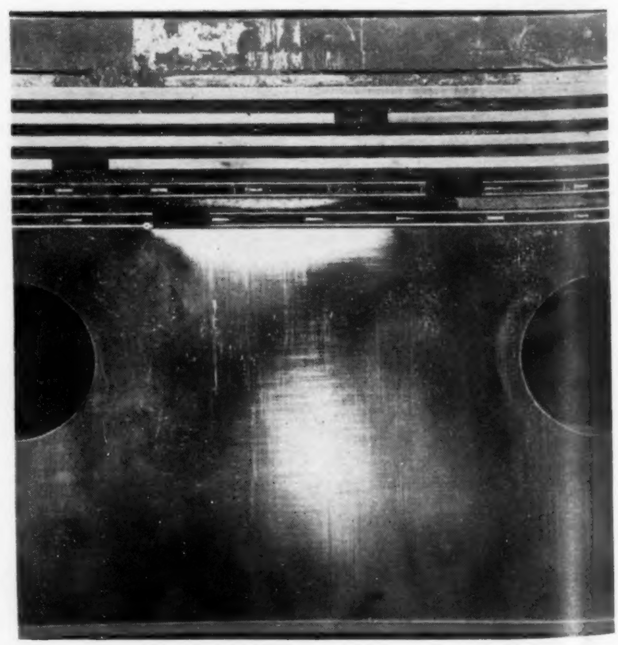
Density and Reflection

The densitometer was developed for use in photographic studies and hence is calibrated in terms of density instead

turn to page 63



■ Fig. 12 – Caterpillar piston after 468 hr under 1-A test conditions using an unsatisfactory lubricant—thrust side



■ Fig. 13 – Caterpillar piston after 468 hr under 1-A test conditions using an unsatisfactory lubricant – antithrust side

COOPERATIVE RESEARCH COMES of AGE

by C. B. VEAL

Manager and Secretary, Cooperative Research Council

THIS year the petroleum and automotive industries signalize the twenty-first anniversary of their joint research by enlarging its sphere, revising its methods, and changing its name.

The interest that has prompted this Section's desire to hear of these new developments, is most natural. One of your former members is generally credited, in conjunction with Henry L. Doherty, then president of Cities Service, with having first conceived the idea of the joint research. Certainly he did more than any other single person to nurture it, by his faith, his enthusiasm, and unselfish devotion. I refer, of course, to Harry L. Horning, who, with his combination of technical acumen and genial, tolerant, understanding of humanity, was the ideal exponent of cooperative enterprise. The project to which, as the Cooperative Fuel Research, he gave generously and tirelessly of his thought and energy, has become now the Cooperative Research Council. We will discuss the development, operation, and, to the extent permitted by military secrecy requirements, the technical projects of the new organism.

The healthy adult stage of the joint research is a natural outgrowth of a lusty and fruitful minority. The Cooperative Fuel Research proved that two among the nation's largest industries can harmoniously join technical forces to overcome common technical difficulties; and that such solutions bring vast economic benefits, not only to the industries themselves, but to the consuming public. It perfected a type of organization for effective and economical joint research. It has built up, within the technical staffs of the two industries, a substantial personnel, educated to the moral obligations and the methods of cooperative effort.

This last-mentioned achievement of the Cooperative Fuel Research, the enlisting and retaining the adherence to it of a thousand or more technical specialists, deserves emphasis. True, we have mourned and deplored the passing of some of its ablest originators. One has been mentioned, Harry Horning. With his name, in this connection, that of Coker Clarkson is inescapably associated. His understanding receptiveness and wise encouragement typify the support the SAE has rendered the Cooperative Fuel Research, to which it has consistently been the largest contributor. However, time has compensated for its depletion of our ranks, bringing substantial reinforcements, and turning raw recruits into seasoned troopers. Many a young man, who five or 15 years ago performed some minor Cooperative Fuel Research job as his first assignment away from the home office, is now one of his company's able executives and a guiding member of a CFR Committee. Only the availability of a large reservoir of trained and loyal workers enables our organization to achieve its present volume of research with the speed necessitated by wartime demands.

IN this paper, Mr. Veal tells the story of the development of the Cooperative Research Council from its creation as the project of three or four men to its growth into an organization of 80 working groups with a total membership of more than a thousand actively interested technical specialists, carrying out its expanded research in 130 participating laboratories.

In its own words, the new Council proposes "to direct cooperative research in developing the best combinations of fuels, lubricants, and equipment powered by internal-combustion engines."

To carry out this goal, a smooth-working organization has been set up. Four main committees assign the work of solving each problem, as it comes up, to an appropriate working group. The men on these groups are interested, qualified technicians who do the actual research work in their suitably equipped laboratories. The titles of the main committees cover the technical scope of the CRC: Cooperative Fuel Research, Cooperative Equipment Research, and Cooperative Lubricants Research. The fourth, the War Advisory Committee, has been set up to deal most expeditiously with the fuels and lubricants problems of the armed forces.

* * *

THE AUTHOR: C. B. VEAL (M '12) was SAE research manager and secretary of the CFR Committee for 16 years prior to his appointment this year as manager and secretary of the Cooperative Research Council. During and after World War I, associated with Charles M. Manly, he was with the Curtiss Aeroplane & Motor Co. in a consulting capacity. Mr. Veal was head of the Machine Design Department of Purdue University for several years immediately preceding World War I, and throughout those years had an extensive consulting practice.

[This paper was presented at the Chicago Section Meeting of the SAE, Chicago, Ill., Nov. 10, 1942.]

To illustrate the importance of experienced judgment in research, let me cite two instances, illustrating different types of defect in research procedure. First, there is the experiment performed by Bob Burns' cousin, Ebenezer, the scientific one. For several days Ebenezer had been observed by his wondering townsfolk busily engaged at the top of a steep cliff overlooking the village. He was strenuously endeavoring to loosen a giant boulder poised on the summit. Finally, it broke loose, and with a crash rolled down the hill, gaining momentum as it went, with Ebenezer in full pursuit. "Don't touch that stone," he shouted, as he followed its destructive course along Main Street. It upset a peddler's cart, scattering pots and pans, crashed into and through the squire's grocery store, toppled the church steeple, shattered the town's largest plate glass, in the bank, and finally fetched up in front of the teller's cage. Ebenezer arrived, breathless from his chase, followed by an alarmed and indignant group of his fellow citizens. He rolled up his sleeves, turned over the huge rock, examined it carefully, and then turned, triumphant with the success of his experiment, and said to his audience, "See, no moss."

This illustrates the type of research that arrives at a correct conclusion, but with an unjustifiable expenditure of energy and material.

Illustrative of another type of research is the story of two carpenters engaged in PWA construction of a building. One was hammering a nail into a wall with great energy, but striking it on the point instead of on the head. Puzzled at his lack of progress, he considered the situation thoughtfully for some time, and then said indignantly to the second carpenter, "who was the fool that put the head on the wrong end of this nail?" The second carpenter stopped his own work, studied his partner's problem from every angle, and then gave his considered opinion. "That's not the trouble; the nail should be hammered into the opposite wall."

This represents a second type of defective research; that which involves little expenditure, but, because of lack of perspicacity, gets nowhere.

Avoiding both these extremes the Cooperative Fuel Research grew from a project with a single immediate aim, conducted by three or four men at the Bureau of Standards, to an organization of 80 working groups with a total membership of more than a thousand actively interested technical specialists, carrying out its expanded research in 90 participating laboratories.

■ Evolution of Cooperative Research Council

Impelled by this growth, the Cooperative Fuel Research in 1937 studied and reconstituted its organization, a prelude to the establishment of the Cooperative Research Council. The Cooperative Fuel Research Steering Committee, as it was organized in 1921, had no permanent chairman, nor any prescribed rules of procedure. It evolved suitable methods for dealing with each new research topic. Young and rapidly enlarging in an untried field, it utilized its freedom from restrictive organization formulas to expand flexibly in the directions indicated by current necessities. Passing beyond the tentative stage, with its bulk threatening unwieldiness, it looked for a more definite organization. In 1937, procedures that had passed the test of practice were codified, and a framework of committees, projects, and working groups was set up.

The next logical step was to apply these methods in other fields in which both industries have a common interest. Lubrication was the most outstanding and the most urgent, since lubrication then required studies characteristic of Cooperative Fuel Research – intensive cooperative effort devoted to current practical problems.

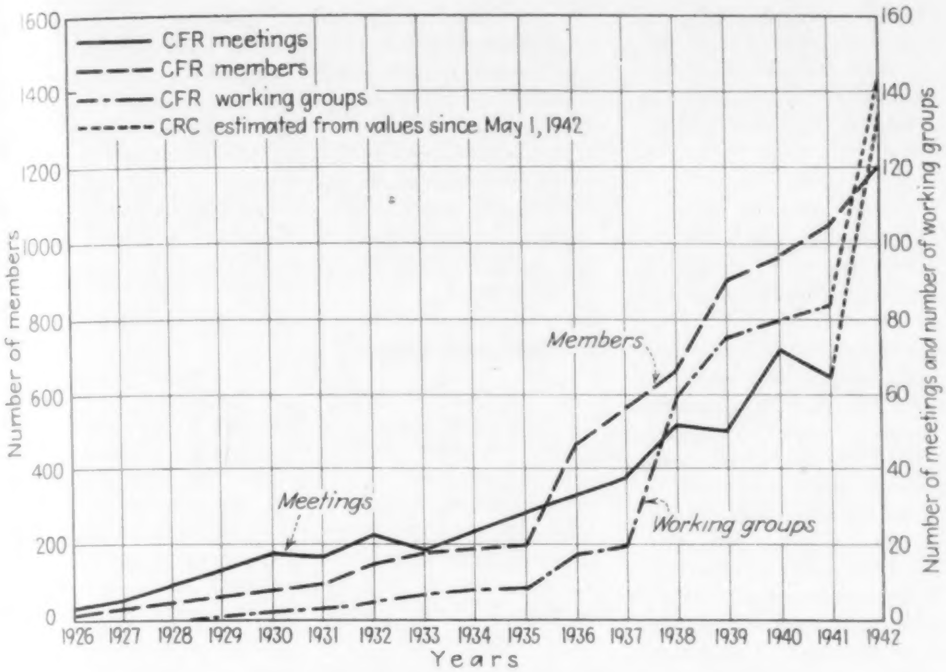
This substantial expansion of the joint research, added to its already notable growth, called for a re-examination of the relationship of the two industries toward it. According to the original understanding, since adhered to, the automotive industry, through the National Automobile Chamber of Commerce, later the Automobile Manufacturers Association, and the petroleum industry, through the American Petroleum Institute, were to bear an equal share of the financial burden. The Society of Automotive Engineers was to maintain the secretariat, as part of the functions of its Research Department. The National Bureau of Standards expected to contribute in services an amount at least equal to the combined contributions of the National Advisory Committee for Aeronautics and the API.

■ Separate Organization Established

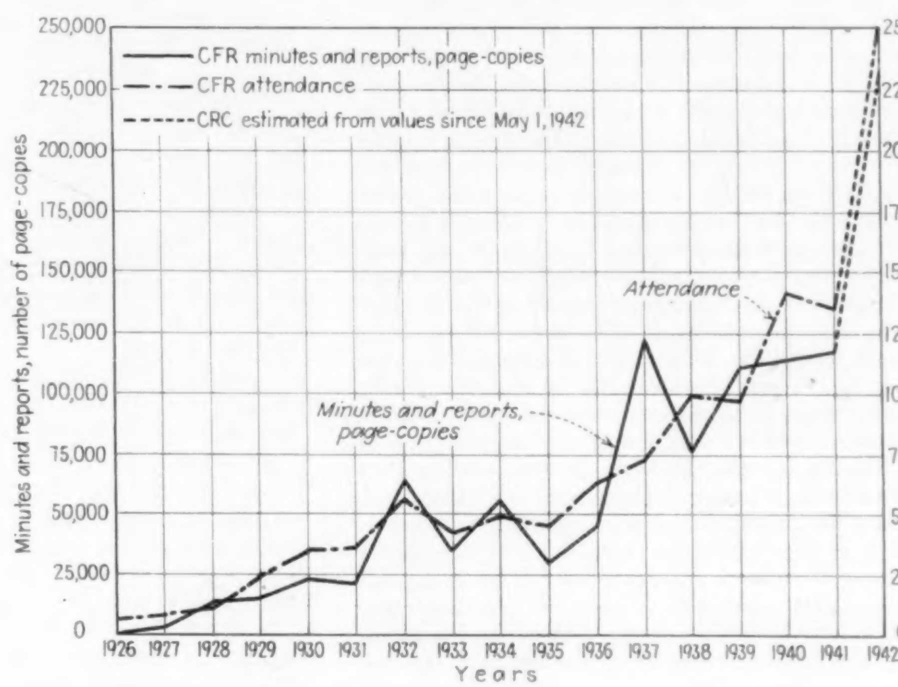
The scope and importance of the joint undertaking now justified its embodiment in an entirely separate organization. It was no longer to function in only one section of the overlapping fields of interest of the petroleum and automotive industries; it was to centralize, correlate, and promote research throughout the entire field. Furthermore, the joint research, vested in a separate body, would have the clarity and definiteness of form advantageous in dealing with other organizations, such as Government bureaus, the Army, and the Navy. This consideration was particularly cogent, since the defense activities of 1940 and 1941 and the direct war effort later brought the armed services more closely and more continuously into contact with the joint research.

A corollary of a separate organization was a separate secretariat. Performing the secretarial functions had entailed for the SAE greater expense than that borne by any other single contributor, a burden which, because of other increased demands, it felt heavily. An illustration of the unforseeable increase in the volume of secretarial work is the production of minutes of committee meetings. A year's activity of the Cooperative Fuel Research Committee, in its early stages, comprised three or four meetings of its 20 or so members, and about 50 pages of mimeograph material. In 1941, CFR committees, with a total membership of 1050, held 63 meetings. In recording and circularizing minutes, 1840 stencils were cut and 115,920 page-copies mimeographed. (See Figs. 1 and 2)

Late in 1941 a special committee of the SAE was appointed to develop ways and means of transferring a part of the cost of the secretariat directly to industry. This group, meeting with a similar committee of the API, formulated a proposal promptly approved by both of the parent organizations. A Cooperative Research Council was to be created, equally representing and supported by the American Petroleum Institute and the Society of Automotive Engineers, and a separate secretariat was to be established. The Cooperative Research Council held its organization meeting on April 13, 1942, and the new secretariat commenced functioning from its office in Rockefeller Plaza, New York, on May 1.



■ Fig. 1 – Growth of CFR: members, meetings, and working groups



■ Fig. 2 – CFR secretariat: production and service record

■ CRC Purpose and Organization

In its own words, the Cooperative Research Council proposes "to direct cooperative research in developing the best combinations of fuels, lubricants, and equipment powered by internal-combustion engines." It has the following self-imposed limitations: "Standardization of methods of test, specifications, and classifications shall not be within the province of the Cooperative Research Council. It is the intent of the Council that such matters shall be promulgated by appropriate existing agencies."

The American Petroleum Institute and the Society of Automotive Engineers govern the Council. Cooperating, in supplying funds or services, are the Automobile Manufacturers Association, the Aeronautical Chamber of Commerce, the National Bureau of Standards, and other technical societies such as the American Society for Testing Materials.

Council membership is 12, appointed for one year, six by each of the two governing bodies. Council members must be empowered to speak authoritatively for their companies on engineering policies, must be active in performing their functions and may not send proxies to Council meetings. Mr. B. B. Bachman is the first chairman and Dr. T. G. Delbridge the first vice-chairman.

The Council's functions are administrative, self-limited

primarily to business phases. It believes that "they govern best who govern least." It prepares and administers budgets and supervises in general all activities and the secretariat. The officers of the latter are a manager and secretary, and a treasurer, Dr. R. P. Anderson, of the American Petroleum Institute.

Technical Organization

In its technical structure, the Council exemplifies the adage that "the child is father to the man." Both in procedure and in the selection of subject matter, it follows the pattern of Cooperative Fuel Research.

Activities are not fitted into a theoretical, *a priori* program. A research project is embarked on only when industry feels its need with sufficient urgency to contribute to it in men, material, and money. When the urgency of a problem has been demonstrated, the appropriate committee or division forms a working group to handle it, by appointing either a group leader, who selects his coworkers, or the group in its entirety. In either case, after the initial stages of its formation, the group becomes autonomous, selecting its own leader and members. These must be interested, qualified technicians who will do the actual research in their suitably equipped laboratories. Group leaders are solely responsible for work under their direction. Working groups are not only the foundation for the entire technical committee structure of the Council; they are also the most important factors in the whole plan for securing prompt and satisfactory research results.

Working groups concerned with allied topics are classified under an appropriate collective title. For each such classification, a project director correlates and guards against overlapping among the projects assigned to him, and the projects assigned to other classifications and under other directors. These classifications are, in turn, grouped into divisions, whose members, group leaders, and project directors, elect their chairmen. Project directors are either appointed by division chairmen or elected by the combined groups whose scope they are to control. Finally, divisions are grouped into four main committees. Members of committees are representatives of the companies in whose laboratories and by whose personnel the projects of the working groups are being carried out. Each main committee has an assignment committee, consisting of the chairman of the main committee, the chairmen of the divisions, and regional representatives. The assignment committees guard against overlapping of effort throughout the entire field sponsored by the Council, and may call upon project directors and group leaders to meet with them.

Three special points must be noted in connection with the procedure and organization outlined. First, due to wartime exigencies, the rule restricting to industry, the initiation of projects has been relaxed. The Ordnance Department, for instance, has requested that the technical committees themselves suggest problems which should be considered, to the end that necessary information may be available before the problem becomes urgent. Also, the secretary, for questions of immediate urgency in relation to the war effort, may get projects under way. Second, in certain instances, the number of laboratories studying a given subject has been so large that to have all of them represented on the working group would result in an unwieldy body. Special measures are taken in such cases to insure equitable representation and effective participation for the interested laboratories. Third, a hierarchy of committees of strictly logical structure and identity of classification and nomenclature throughout is not sought. For instance, in the Motor Fuels Division, vapor lock is part of the volatility project; in the Aviation Fuels Division, it is a separate project coordinate with volatility. Factors in deciding the status of a project are: the personnel involved, the organization of the industry interested, the importance and urgency of the subject in relation to existing projects. "Get the job done," not "follow the system," is the principle.

Technical Scope

The titles of the Council's four main committees cover its technical scope. The Cooperative Fuel Research Committee has come under its sponsorship without change. The five divisions of this Committee deal with motor, aviation, automotive-diesel, and non-petroleum fuels, and with the gasoline survey. The Cooperative Lubricants Research Committee has taken over the programs of the following former SAE groups: Petroleum Group of subdivision B of the Lubricants Division of the Standards Committee, Aircraft-Engine Lubricants Research Committee, Crankcase Oil Stability Research Committee, Engine Wear Research Committee, and the Extreme-Pressure Lubricants Research Committee. Its three divisions are: Engine Oil, including projects on performance and test methods; General, dealing with chassis, transmission and gear lubricants, and hydraulic fluids; and Lubrication, for the study of the fundamentals of lubrication theory and practice. The Cooperative Equipment Research Committee is concerned with mechanical design features, of either engine or vehicle, involved in the usage of fuels or lubricants. This committee will study the problems and present the viewpoints of oil men, equipment manufacturers, and consumers or operators. The fourth committee, the War Advisory Committee, is, in membership and functions, the former CFR-SAE Fuels and Lubricants Advisory Committee. It maintains contact with our military services and other Government agencies.

An adequate and accurate summation of the volume of research sponsored by the Council would require a time-consuming census, particularly inappropriate in these urgent days, of the 130 participating laboratories. This volume, and its probable increase, are reflected in a few figures readily available at the secretariat. Since May 1, 1942, when the secretariat commenced functioning, 58 committee meetings with a total attendance of 1109 have been held. In reporting and circularizing minutes of these meetings, 1400 stencils have been cut and 100,000 page-copies processed. That the pace of activity for the first five months of the Council's existence will be maintained is a conservative supposition. Nevertheless, on this hypothesis, as reflected in committee meetings and the actions taken at them, the Council's yearly research activity will be 100% greater than that of the CFR and 50% greater than the combined activity of all committees whose programs it assumed.

Descriptions of projects and reports of results must be carefully circumscribed, since practically all of our current research pertains to our armed services. However, it is no military secret that our airplanes, tanks, motor trucks, and guns are being used in the tropical climate of the South Seas and the Libyan Desert, and in the sub-zero

temperatures of Alaska and the steppes of Russia. Our research is concerned with all these types of equipment and its operation over these wide temperature ranges.

An idea as to the volume of the research may also be obtained from the organization chart, Fig. 3, which shows the organization of the committees and divisions. Fig. 4 shows the organization of the Aviation Fuels Division – a typical division of CRC. Even a bare enumeration of the projects would be tiresomely time-consuming.

Examples of Current Activities

A few details concerning the current activities of each of the four main committees may adequately indicate the nature of the research sponsored by the Council.

The Cooperative Lubricants Research Committee, since May 1, 1942, has organized on the CFR pattern the activities of the bodies whose functions it assumed. It has more than 20 active projects in its three divisions, and has already released its first major report, the results of a cooperative program of laboratory bench tests on engine oils. This report was presented as a paper at the SAE National Fuels and Lubricants Meeting at Tulsa, Oct. 22 and 23, 1942.

On the current program of the Cooperative Equipment Research Committee an item of pertinent interest is the 1942 Motor Survey. Seven companies contributed data to this survey and a report based on these data has been prepared by the Bureau of Standards. In former years the purpose of these surveys has been the determination of the fuel quality requirements of automobiles with respect to vapor lock and knock. The current survey has been adapted to present conditions by providing information on the need and extent of changes or adjustments which automotive equipment may require to operate satisfactorily on fuels of lower octane number and impaired volatility, and by investigating the resultant effects of such changes or adjustments on power, economy, maintenance, and other operating characteristics.

The Cooperative Fuel Research Committee, firmly established and in full operation when it came under the sponsorship of the Cooperative Research Council, has a great number of active projects. Four of these, on which reports of major importance have been issued since May 1, 1942, will serve as illustrations.

The Motor Fuels Division issued its third triennial analysis of the Precision of Motor Fuel Testing. This was based on 7000 knock ratings on 183 fuels and on inspection data on 109 of these fuels. The knock ratings were obtained from monthly Motor Fuels Exchange Tests, in which 19 laboratories participate, and from semi-annual exchange tests, in which about 70 laboratories participate. In addition to its evaluation of knock testing, the report also analyzed the precision of measurements of vapor pressure, gravity, and distillation temperatures.

During July, 1942, the road test group of the Motor Fuels Division participated in a series of tests under the auspices of the Quartermaster Department to determine the octane requirements and vapor locking characteristics of Quartermaster Corps Vehicles in desert operation. The work of the group, conducted at Camp Seeley, Calif., was covered in a 58-page confidential report to the Quartermaster.

The Automotive Diesel Fuels Division recently completed a series of full-scale engine tests to determine the effect of cetane number, volatility, gravity, and viscosity on five factors of engine operation: (1) engine deposits, (2) odor and lachrymation, (3) ignition quality under low-temperature starting conditions, (4) power output, fuel consumption, exhaust cleanliness, and (5) engine smoothness. Seven fuels were so selected as to permit the variation of a single physical property while all other properties remained constant. A 55-page analysis report on these data has been completed and is now undergoing editorial revision.

A confidential report to the Aeronautical Board on the detonation rating of fighting grade aviation fuel has been made by the Aviation Fuels Division. This was based on full-scale engine ratings of 14 fuels under takeoff and simulated cruising power conditions. Other detonation projects are full-scale single-cylinder tests and two monthly exchange tests of the laboratory groups. Extensive programs on volatility and vapor lock, involving expenditures running well into six figures, have been prepared by the Aviation Fuels Division as the basis for contracts with the Army Air Forces and the Navy Bureau of Aeronautics.

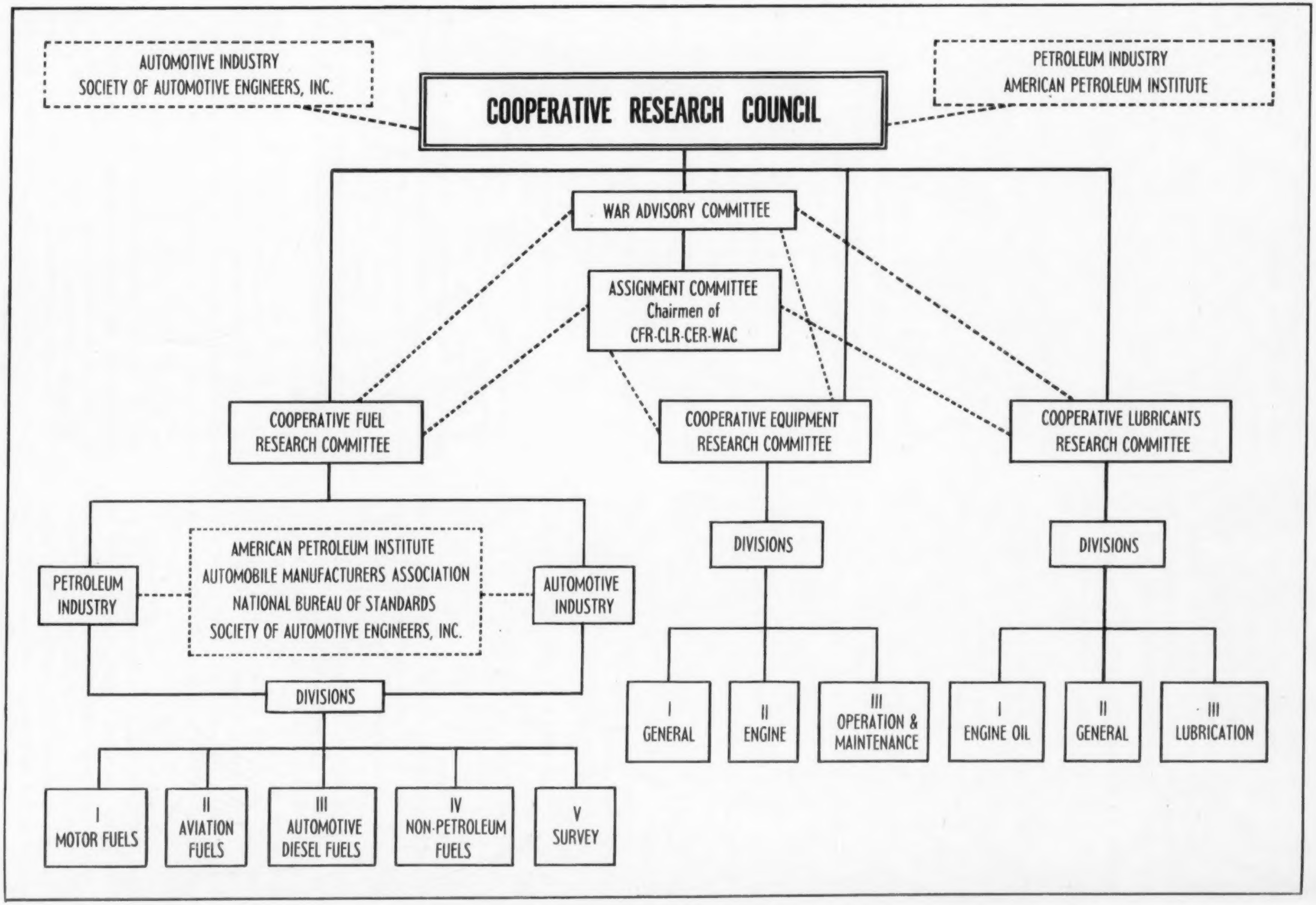
Some Wartime Projects

In general, major interest naturally centers at this time on the activities of the War Advisory Committee. The problems of the armed services are presented to the Council through the War Advisory Committee. Two exceptions must be noted; both are aimed at using existing facilities most expeditiously. All projects on aviation fuels come directly under the province of the Aviation Fuels Division, which maintains contact with the Army Air Forces and the Navy Bureau of Aeronautics. The Aviation Fuels Division has behind it a long history of such activity, extending back to pre-war emergency days, and so has had available the experience and machinery for handling quickly and effectively the fuels problems of wartime aviation. The second exception is that a subject may be brought directly to the attention of a committee, division, or working group, where such a body is working on an allied topic, or includes in its membership a representative of the service branch directly interested in the problem.

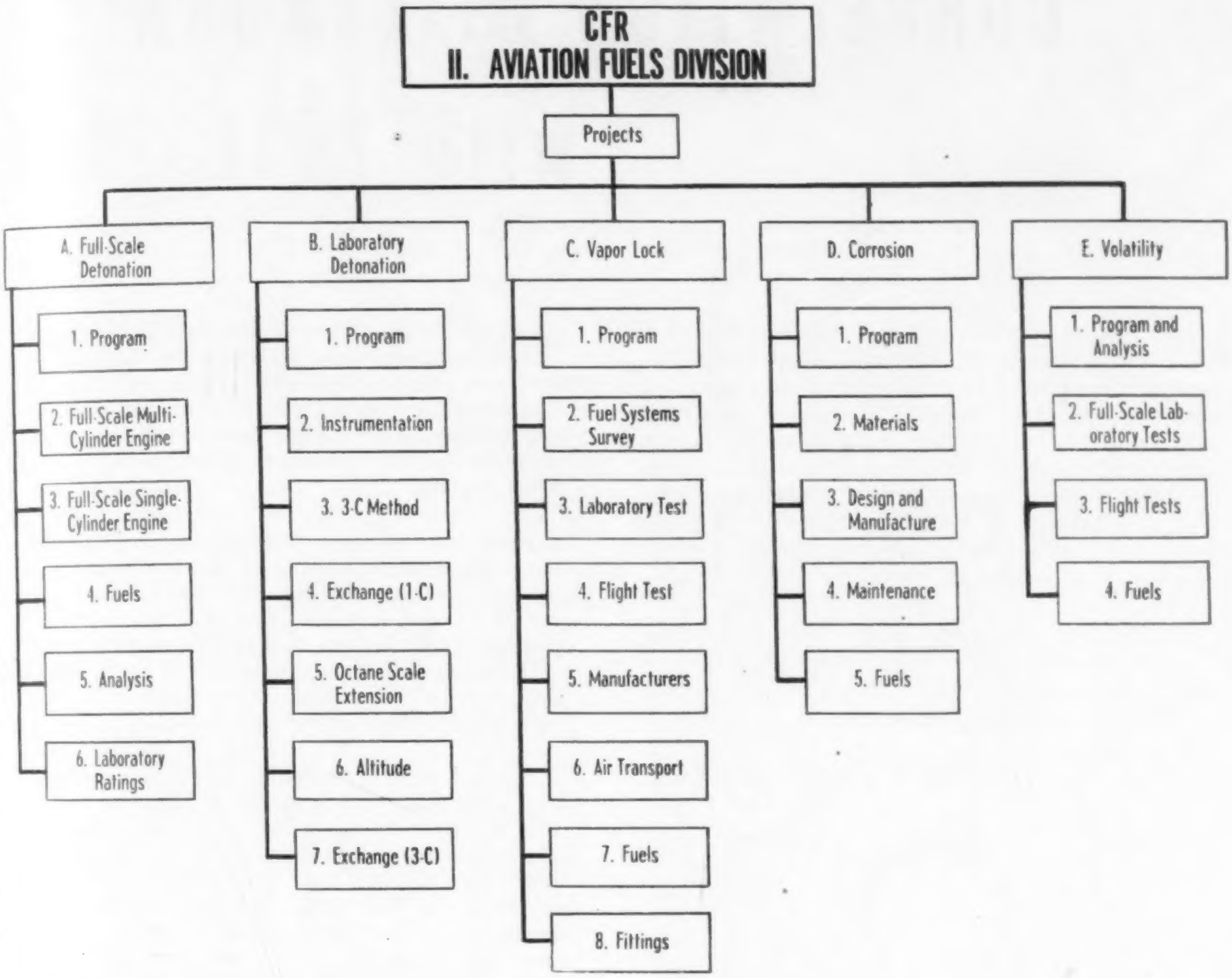
Official recognition is accorded the War Advisory Committee or other main committees, in all matters of fuels and lubricants and their military utilization, by the following bodies: War Department Committee on Liquid Fuels and Lubricants, Ordnance Department, Quartermaster Corps, Corps of Engineers, Chemical Warfare, Signal Corps, Army Air Forces, Navy Bureau of Aeronautics, National Bureau of Standards, Bureau of Mines, Bureau of Economic Warfare, Office of Defense Transportation, the British Air Ministry, and the National Research Council of Canada.

When any of these bodies places a question before the War Advisory Committee, the latter immediately takes such action as will bring quickest results. If an appropriate working group is already in existence, the project is assigned to it. If not, the War Advisory Committee forms the group, or authorizes the secretary to do so. This flexibility of procedure has so speeded up the organization of research as, in some instances, to shorten to a single working day the period between the putting of the question and the start of actual investigation. After such a project has gotten under way it is assigned to the appropriate committee and division, so that the War Advisory Committee may keep itself free for its primary functions of contact and administration.

Sixty-eight questions put by the armed services have



■ Fig. 3 – Organization chart for the Cooperative Research Council



■ Fig. 4 – Organization chart for the CFR Aviation Fuels Division – a typical division of CRC

been considered and have had recommendations made concerning them by the War Advisory Committee. They have necessitated research and experimental work in 38 projects: 15 concerned with crankcase oil and engine lubrication, 12 with transmission and gear lubrication, five with other lubrication problems, and six with fuel and its utilization.

Within the realm of engine crankcase lubrication the following subjects have been included: oil filters; air cleaners; engine wear; engine-oil characteristics; used engine oils, including reclaiming, re-refining and test procedures for their evaluations; cold starting, including priming systems, dilution, heaters, and oil pumpability; specifications and acceptance tests for both heavy- and moderate-duty oils; top cylinder lubricants and compatibility of specification oils.

Transmission and gear lubrication has called for study on the following phases: temperature effects on various grades of lubricant; requirements for operation under both extremely low- and extremely high-temperature conditions, and corrosion in the presence of water.

Of particular current interest are the investigations on grease. Cooperative research on greases was practically non-existent until war requirements called for a great amount of information that could be obtained only experimentally. First one, and later two, working groups cooperating with the laboratories of the three arsenals undertook intensive programs dealing with test methods for stability, low-temperature evaluation, pumpability and water resistance of greases, studies of greases for recoil and equilibrator mechanisms, rust-inhibiting greases, and lubrication of high-speed artillery carriages.

Miscellaneous lubrication studies are concerned with specifications for wheel bearing greases, stability test methods for torque and converter fluids, relative merits of grease and oil for universal joints, and the lubrication of ferrous metal combinations substituted for critical non-ferrous metals.

In the field of fuels, the War Advisory Committee has initiated projects on the effect of sulfur in gasoline, special fuel additives, the octane requirements of military vehicles, and vapor-pressure tolerance of military vehicles, including tanks.

Of civilian as well as military interest is the study being

turn to page 63

CORRELATION of LABORATORY OI[L TESTS]

with FULL-SCALE EN[GINE TESTS]

TWENTY laboratories, cooperating in the Bench Test Subcommittee's program, ran tests on six reference oils in Underwood apparatus, Lauson single-cylinder engines, and in a number of other types of laboratory oil test devices. The six reference oils had an extensive test background in full-scale engines.

Agreement among laboratories testing the same oils in the same type of apparatus was surprisingly good considering that standardized test conditions were not used and that the test procedures used by the cooperating laboratories varied considerably. Correlation of the laboratory tests with full-scale engine tests was also very good in the majority of instances.

The test data compiled from the cooperative work emphasize the importance of oxidation catalysts and test temperatures on oil deterioration and the degree of correlation between laboratory bench tests and full-scale engine tests. The cooperative test data also indicate suitable test conditions in the various types of bench apparatus which tend to produce the most consistent results indicative of full-scale engine performance as well as indicating inadequate or extreme test conditions which may tend to give misleading results.

AT the January 1942 meeting, J. B. Fisher, chairman of the SAE Crankcase Oil Stability Research Committee, appointed the Bench Test Subcommittee[1] to carry out a cooperative bench test program, involving the testing of six reference oils in the Underwood apparatus, Lauson single-cylinder engines, and in bench test apparatus of other types, as selected by cooperating laboratories.

Following the organization of the Cooperative Research Council under the joint sponsorship of the SAE and the API, the various oil research committees of the SAE were transferred to the new Council.

For the benefit of those not completely acquainted with this change, following is the new organization set-up:

Cooperative Research Council – C. B. Veal, secretary

Cooperative Lubricants Research Committee – C. G. A. Rosen, chairman

Engine Oils Research Division – J. B. Macauley, chairman

Test Methods Projects – H. C. Mougey, director

Under the new organization, the Bench Test Subcommittee accordingly functions as a part of the Test Methods Group under the direction of Mr. Mougey.

The work described in this report was done by 20 cooperating laboratories[2]. This paper is not a formal report of the Bench Test Subcommittee, but is a compilation of the data secured by the cooperating laboratories and an interpretation of same as prepared by the author.

The six reference oils used in the cooperative work were the B-1, B-2, and B-3 oils from Subdivision B of the SAE Lubricants Division, as used in standardizing the 36-hr Chevrolet engine oil stability test (now the 36-Hr Oxidation Test); and the C-1, C-2, and C-3 oils from the CFR Fuels and Lubricants Advisory Committee, Chevrolet Engine Test Procedure Group. All six of these oils accordingly have an extensive background of Chevrolet engine tests.

The objectives of the Bench Test Subcommittee were outlined as follows:

1. To determine the degree of correlation of the various bench tests with Chevrolet engine tests.

2. To determine the agreement among the different laboratories testing the same oils in the same apparatus.

3. To determine the effect of Underwood and Lauson engine test conditions on correlation with Chevrolet engine tests and on agreement among the different laboratories.

In the cooperative work, each laboratory was requested to test the six oils following the test procedures each had used in the past. Standardized test conditions were not set up for any of the bench test methods, so this work was not a standardization program but merely an exploratory program. The results of this cooperative work should, therefore, be viewed accordingly.

Reference Oils

Table 1 is a summary of the general properties of the six reference oils used in the cooperative test work. These six oils represent a good cross-section of the various types and varieties of motor oils on the market, ranging from oils of moderately low V. I. to quite high V. I., and from oils

[This paper was presented at the National Fuels and Lubricants Meeting of the SAE, Tulsa, Okla., Oct. 23, 1942.]

[1] *Bench Test Subcommittee:* C. W. Georgi, chairman, H. A. Ambrose, E. R. Barnard, F. C. Burk, A. G. Cattaneo, J. B. Fisher, Raymond Haskell, Hugh Hemmingway, F. W. Kavanagh, F. L. Miller, H. L. Moir, C. C. Moore, H. C. Mougey, T. G. Murphy, R. I. Potter, J. R. Sabina, B. E. Sibley, C. E. Stevenson, A. O. Willey, D. B. Williams.

[2] *Cooperating Laboratories:* Atlantic Refining Co., Carbide & Carbon Chemicals Corp., Continental Oil Co., E. I. du Pont de Nemours & Co., Inc., Franklin Creek Refining Corp., General Motors Corp., Gulf Research and Development Co., Kendall Refining Co., Lubri-Zol Corp., Pennsylvania State College, The Pure Oil Co., Quaker State Oil Refining Corp., Shell Development Co., Standard Oil Co. of Calif., Standard Oil Co. of Indiana, Standard Oil Co. of Ohio, Standard Oil Development Co., The Texas Co., Union Oil Co., Waukesha Motor Co.

OIL BENCH TESTS
ENGINE TESTS

by C. W. GEORGI
Technical Director,
Quaker State Oil Refining Corp.

containing no additives to those containing both inhibitor and detergent addition agents.

Fig. 1 shows typical pistons from Chevrolet engine tests on the "B" oils using the ASTM Proposed Method of Test (280 F crankcase oil temperature). In the case of oil B-1 the pistons were very clean at the end of 36 test hr, but showed considerable lacquer accumulation at the end of 67 test hr. With oil B-2 the pistons remained clean and substantially lacquer-free but the test had to be shut down at the end of 32 hr because of excessive copper-lead bearing corrosion. Oil B-3 left clean pistons at the end of both 36- and 67-hr tests.

Fig. 2 shows typical pistons from Chevrolet engine tests on the "C" oils. Oil C-1 left clean and substantially lacquer-free pistons after 36 test hr following both the ASTM Proposed Method with 280 F crankcase oil temperature and the modified "C" procedure with a 275 F crankcase oil temperâture. Oil C-2 also left quite clean pistons after the 36-hr test under the "C" procedure with 275 F oil temperature. Oil C-3 developed very heavy piston lacquer after 36 hr with 275 F crankcase oil temperature and still showed quite heavy lacquer formation after a 36-hr test with a reduced crankcase oil temperature of 265 F.

Figs. 3 and 4 show push-rod cover plates from the same Chevrolet tests. Sludge deposition characteristics follow the same general order as the respective piston conditions for the six reference oils.

These photographs are from Chevrolet tests conducted by one laboratory on the six oils, but since the general agreement among laboratories in running cooperative Chevrolet engine tests has been very good, it is believed these are typical results and give a general picture of the respective varnish and sludge ratings of the six oils in the Chevrolet engine oil stability tests.

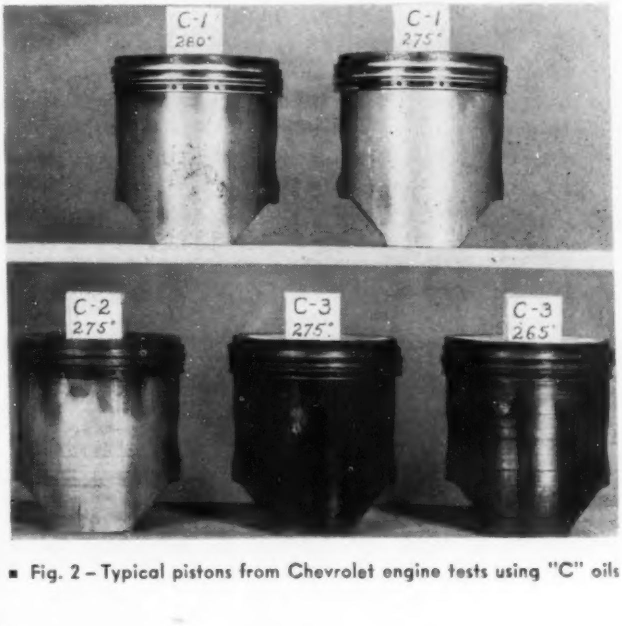
Table 2 shows copper-lead bearing corrosion data from 36-hr Chevrolet engine tests. Oils B-1, B-3, C-1, and C-3 are rated as being substantially non-corrosive to copper-lead bearings, while oils B-2 and C-2 are rated as highly corrosive.

Table 3 shows sludge formation characteristics of the six reference oils after 36-hr Chevrolet tests, as measured by the drain-oil analysis for naphtha insolubles and chloroform solubles. Oils B-1 and C-3 are rated as having a high rate

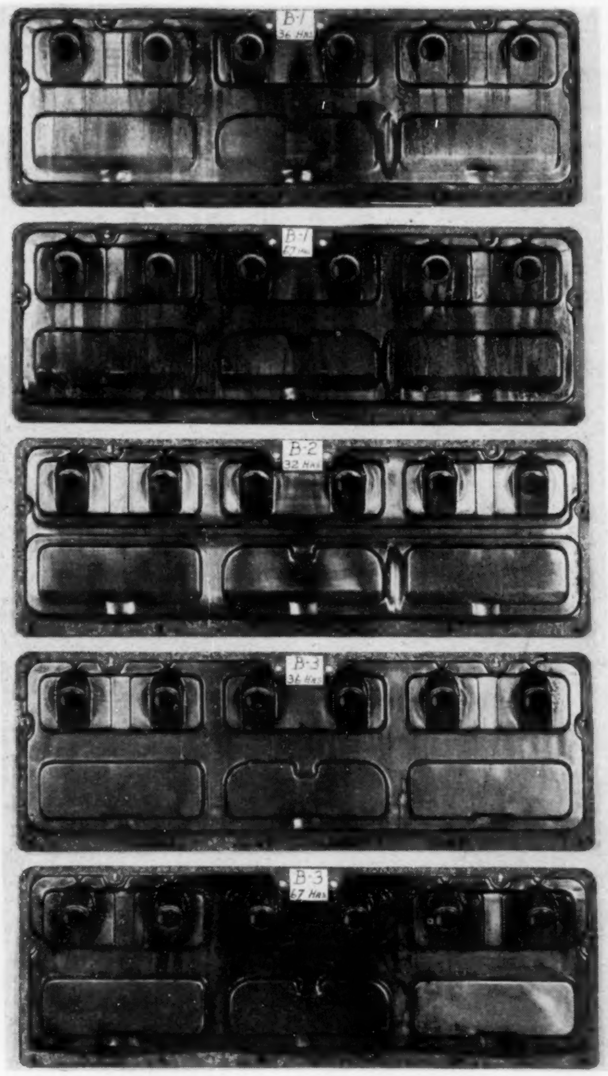
Table 1 – **Reference Oils Used in Cooperative Bench Tests**

B-1	57-V. I. oil, containing inhibitor and metal detergent additives
B-2	106-V. I. oil, no additives
B-3	100-V. I. oil, containing inhibitor and metal detergent additives
C-1	88-V. I. oil, containing sulfur inhibitor compound
C-2	92-V. I. oil, containing phosphorus inhibitor compound
C-3	76-V. I. oil with natural sulfur content of 0.37%, no additives

■ Fig. 1 – Typical pistons from Chevrolet engine tests using "B" oils



■ Fig. 2 – Typical pistons from Chevrolet engine tests using "C" oils



■ Fig. 3 – Push-rod cover plates – from Chevrolet engine tests using "B" oils

of sludge formation and the other four oils as being quite resistant to this form of oxidation.

Since a number of the cooperative bench tests were run in single-cylinder Lauson engines under conditions intended to evaluate oil detergency, the detergency properties of the reference oils are of interest. Fig. 5 shows single-cylinder diesel engine pistons after 480-hr test runs. Oils B-1 and B-3 show no evidence of piston-skirt lacquer formation and the ring zones show the desired condition of

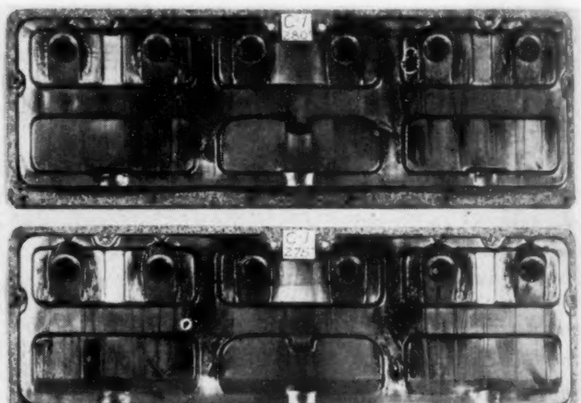
Table 2 – Copper-Lead Bearing Corrosion 36-hr Chevrolet Tests

Oil No.	Loss Per Half Bearing, g	Remarks
B-1	0.325	Average of 8 laboratories "B" procedure – 280 F oil temperature
B-2	1.80	Average of 8 laboratories "B" procedure – 280 F oil temperature
B-3	0.271	Average of 8 laboratories "B" procedure – 280 F oil temperature
C-1	0.086	Average of 14 laboratories "C" procedure – 275 F oil temperature
C-2	1.50	Average of 14 laboratories "C" procedure – 275 F oil temperature
C-3	0.168	Average of 13 laboratories "C" procedure – 275 F oil temperature

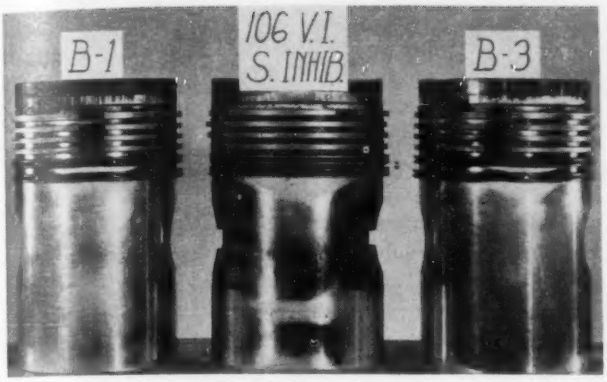
Table 3 - Drain-Oil Inspection - 36-hr Chevrolet Tests

Oil No.	Naphtha Insoluble, %	Chloroform Soluble, %	Remarks
B-1	1.72	1.30	Average of 8 laboratories "B" procedure – 280 F oil temperature
B-2	0.71	0.192	Average of 8 laboratories "B" procedure – 280 F oil temperature
B-3	0.50	0.10	Average of 8 laboratories "B" procedure – 280 F oil temperature
C-1	0.81	0.13	Average of 14 laboratories "C" procedure – 275 F oil temperature
C-2	0.68	0.18	Average of 14 laboratories "C" procedure – 275 F oil temperature
C-3	1.98	1.52	Average of 13 laboratories "C" procedure – 275 F oil temperature

cleanliness and freedom from ring sticking. The center piston in Fig. 5 is from a 480-hr test on an oil of 106 V.I., quite similar to oil B-2, but containing a sulfur-type inhibitor similar to that in oil C-1. 36-hr Chevrolet tests on this oil were very similar to oil C-1. As shown, this piston has heavy varnish deposits on the skirt and heavy deposits in the oil rings, ring grooves, and lands. The top and



■ Fig. 4 – Push-rod cover plates – from Chevrolet engine tests using "C" oils



▪ Fig. 5 – Single-cylinder diesel engine pistons after 480-hr test runs

Table 4 – Summary of Chevrolet Engine Test Ratings of Reference Oils

Oil No.	Oil Stability	Engine Cleanliness	Bearing Corrosion	Detergency[1]
B-1	Fair to good	Fair to good	Non-corrosive	Good
B-2	Poor (Acid Formation)	Good	Corrosive	Poor
B-3	Good	Good	Non-corrosive	Good
C-1	Good	Good	Non-corrosive	Limited
C-2	Fair (Acid Formation)	Good	Corrosive	Poor
C-3	Poor (Sludge Formation)	Poor	Non-corrosive	Poor

[1] Based on available data from diesel engine tests.

fourth compression rings and the bottom oil ring were also partially stuck.

Table 4 is a summary of the Chevrolet engine test ratings of the six reference oils. Included also is an estimate of their detergency properties based on the information available.

A wide variety of oil types is accordingly represented in the six reference oils, and it seems reasonable to assume that any bench test apparatus which will rate the six oils in their proper order will have promise of considerable usefulness.

▪ Underwood Tests

The cooperating laboratories which ran Underwood tests reported use of a considerable variety of test conditions:

2 Laboratories reported tests at 275 F with 0.01% Fe_2O_3 (as naphthenate) catalyst.

1 Laboratory reported tests at 325 F with 0.005% Fe_2O_3 catalyst.

9 Laboratories reported tests at 325 F with 0.01% Fe_2O_3 catalyst.

4 Laboratories reported tests at 325 F without catalyst.

5 Laboratories ran tests with cadmium bearings only.

4 Laboratories ran tests with copper-lead bearings only.

6 Laboratories ran tests with both cadmium and copper-lead bearings.

Four different makes of cadmium bearings and three different makes of copper-lead bearings were reported.

All laboratories but one reported use of the polished 2 x 10 in. copper baffle.

11 Laboratories used 1500-cc oil samples.

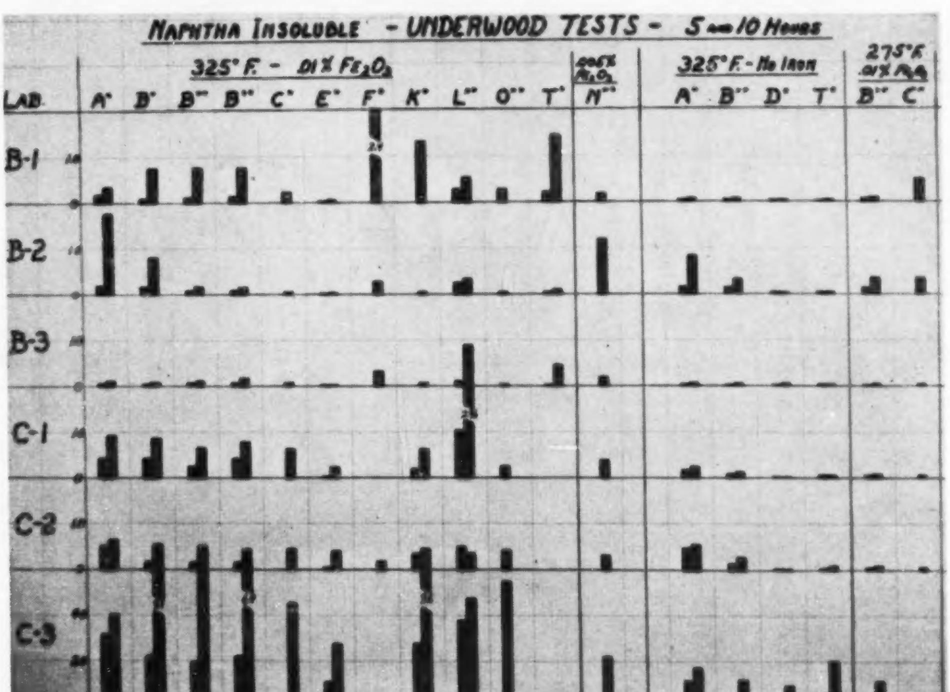
1 Laboratory used 1600-g oil samples.

1 Laboratory used 2.8-lb oil samples.

8 Laboratories used the solvent cleaning method between test runs as recommended in the original Underwood test procedure.

6 Laboratories used thorough cleaning between tests, involving boiling alkali solutions or the like.

11 Laboratories used 10 psi oil pressure.



▪ Fig. 6 – Naphtha insoluble data – Underwood tests

1 Laboratory used 5 to 7 psi oil pressure.

1 Laboratory used 15 psi oil pressure.

Fig. 6 shows the naphtha insoluble values from the cooperative Underwood tests. (In the tables, the laboratories marked with a single asterisk used the solvent cleaning method, and the laboratories marked with a double asterisk used thorough cleaning between test runs.)

It will be noted that the naphtha-insoluble results are in good general agreement, using the 325 F test temperature and 0.01% Fe_2O_3 (as iron naphthenate) catalyst, considering the many variations in individual test procedure details. The Underwood runs also rate the six oils in the same order as the Chevrolet stability tests. Laboratories C, E, and O show lower results than the other laboratories. Since these three laboratories report results lower than the average with all six oils, some variable in test conditions is probably the cause.

Laboratory B ran Underwood tests using both solvent cleaning and thorough cleaning with alkali solution between test runs. Check tests were also made on two different Underwood machines. There is very good agreement on these three sets of results. Laboratory N ran Underwood tests at 325 F with 0.005% Fe_2O_3 catalyst. The naphtha-insoluble formation under these conditions is appreciably lower than the general average of the tests with 0.01% Fe_2O_3 catalyst, and indicates the importance of uniform catalyst and catalyst concentration.

Tests by four laboratories at 325 F with no iron catalyst tend to rate the oils quite close together as to sludge formation, and it would appear that use of soluble iron catalyst is necessary to obtain a proper spread between the oils, in line with Chevrolet engine tests.

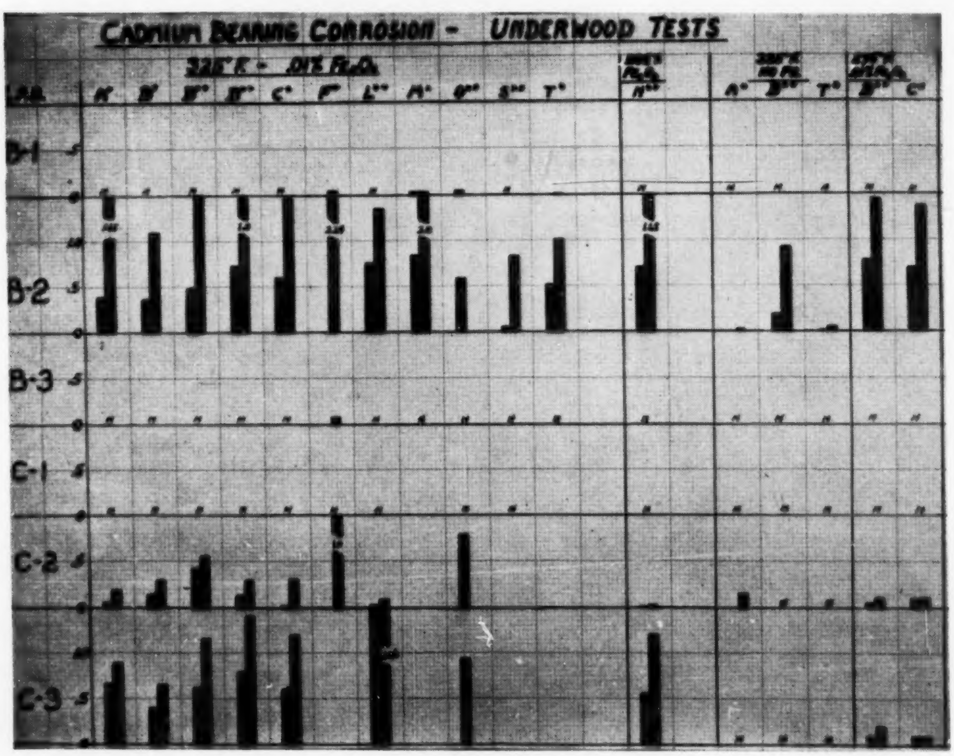
Two laboratories ran tests at 275 F with 0.01% Fe_2O_3 catalyst. The naphtha insoluble results under these conditions are very similar to the results at 325 F without iron catalyst. It would appear, therefore, that 0.01% Fe_2O_3 in the form of iron naphthenate, is roughly the equivalent of 50 deg in test temperature as far as sludge formation is concerned.

Taking into consideration the variety of test procedures used by the cooperating laboratories, the naphtha insoluble data are in very good general agreement, and those cases where individual laboratories reported appreciable deviation from the general average can very probably be accounted for by some such variations.

Fig. 7 shows cadmium bearing corrosion data from the cooperative Underwood tests, and serves to illustrate the importance of test temperature and iron catalyst concentration. The laboratories using 325 F test temperature and 0.01% Fe_2O_3 catalyst are in remarkably good agreement, considering the many variations in test procedure details and the different makes of cadmium bearings used. The Underwood apparatus with 325 F test temperature and 0.01% Fe_2O_3 catalyst rates the oils in the same order of corrosiveness as the Chevrolet engine tests, with the single exception of oil C-3, which is rated as highly corrosive by the Underwoods but was definitely non-corrosive in 36-hr Chevrolet stability tests.

Laboratory N, which used 0.005% iron catalyst and 325 F test temperature, underrates the corrosiveness of oil C-2 compared to the Chevrolet tests, but this milder catalyst-addition still rates oil C-3 as corrosive.

The laboratories running Underwoods at 325 F without iron catalyst are in rather poor agreement on oil B-2, two



■ Fig. 7 – Cadmium bearing corrosion data – Underwood tests

of the laboratories rating this oil as non-corrosive, the other as corrosive. Oil C-2 is also rated as substantially non-corrosive by these test conditions, which is contrary to the Chevrolet ratings. While the elimination of the iron catalyst brings the corrosion rating of oil C-3 into line with the Chevrolet rating, this procedure apparently errs in mis-rating oils B-2 and C-2.

The corrosion data from the runs at 275 F with 0.01% iron catalyst rate C-3 as substantially non-corrosive, in line with the Chevrolet ratings, but also rate oil C-2 as non-corrosive, contrary to the Chevrolet ratings.

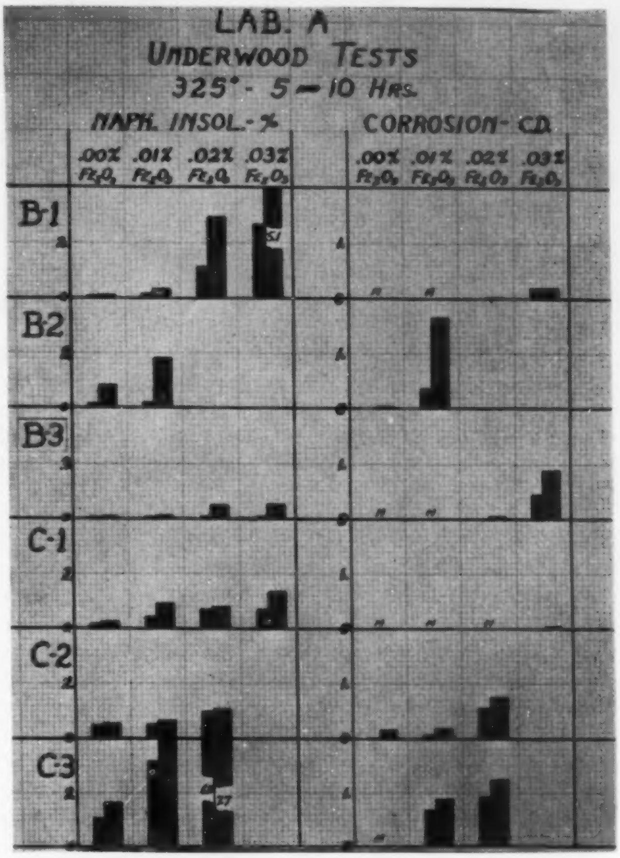
The cadmium bearing corrosion data indicate that 10-hr Underwood tests at 325 F, with 0.01% Fe_2O_3 as naphthenate, come the closest in rating the corrosion tendencies of the six reference oils in line with the 36-hr Chevrolet tests. The fact that these Underwood test conditions rate oil C-3 as corrosive may not be too critical, since both the Underwood and the Chevrolet engine rate oil C-3 as being unstable both as to sludge formation and acid formation. The fundamental difference seems to be that the oxidation acids formed are not corrosive to sensitive bearings in the engine but are active on bearings in the laboratory test. It may be that oils of the C-3 type form protective varnish coatings on bearings in engines but that such protective deposits do not have an opportunity to form in the Underwood.

Table 5 shows a summary of the copper-lead bearing corrosion data from the cooperative Underwood tests. There is a complete lack of correlation between Underwood copper-lead bearing corrosion and the Chevrolet results. There is also a wide variation in the corrosion data reported by the different laboratories which utilized copper-lead bearings in their Underwoods. The oils containing sulfur additives tend to show high rates of copper-lead bearing weight loss, which is probably due to the fact that during test the sulfur additives formed sulfide films on the bearing surfaces which may have been washed off by the oil blast during test or by the cleaning of the test bearings prior to reweighing at the end of the test. In any event, the high copper-lead weight losses reported by several laboratories with oils containing sulfur inhibitors are directly contrary to the Chevrolet ratings.

Table 5 – Copper-Lead Bearing Corrosion— Cooperative Underwood Tests

Oil	Loss, g
B-1	0.000 to 0.177
B-2	0.05 to 0.240
B-3	0.000 to 0.220
C-1	0.000 to 0.215
C-2	0.000 to 0.05
C-3	0.010 to 0.084

Many investigators have pointed out the difficulty of correlating laboratory tests with engine tests on copper-lead bearing corrosion, which may be largely due to the absence of the wiping action of a shaft on loaded bearing surfaces in most laboratory procedures. The complete lack of agreement of copper-lead bearing corrosion data in the Underwood tests would indicate that cadmium bearing specimens must be used to secure information on corrosion tendencies of oils. In fact, copper-lead bearing corrosion data from Underwood tests may be quite misleading.



■ Fig. 8 – Underwood test results – Laboratory A

Many laboratories reported only partial oil-analysis data from their Underwood runs so that oil deterioration values of viscosity increase, acid number, and carbon residue increase were not tabulated. However, these data, when reported, were of the same general order as the naphtha insoluble and cadmium bearing corrosion results, and follow the same general condition of surprisingly good agreement considering the many variations in test procedure details.

■ Varying Concentrations of Iron Catalyst

Laboratory A ran Underwood tests with increasing concentrations of iron catalyst to secure additional information on the oxidation resistance characteristics of oils and to determine the type of oxidation products formed when highly stable oils are finally subjected to sufficiently severe conditions to cause definite deterioration. Fig. 8 summarizes the data supplied by laboratory A. In the case of oil B-1, increasing concentrations of iron catalyst cause a rapid increase in the amount of naphtha insolubles formed, although the oil remains substantially non-corrosive up to 0.03% Fe_2O_3. Oil B-2 possesses little oxidation resistance and develops both high naphtha insolubles and high bearing corrosion with only 0.01% Fe_2O_3. Oil B-3 is very resistant to oxidation with concentrations of iron catalyst up to 0.03% but with higher concentrations it becomes corrosive, although the sludge forming tendencies still remains low. Oil C-1 retains its oxidation resistance both as to sludge formation and corrosion tendencies with iron concentrations as high as 0.03% Fe_2O_3. Oil C-2

shows moderate increases in corrosiveness and sludge formation tendencies with increasing catalyst additions; and oil C-3, being generally unstable, deteriorates rapidly with respect to both sludge formation and acid formation.

In making Underwood tests following this procedure, it is possible to differentiate between oils of the four basic types of stability.

1. Oils which deteriorate rapidly as to both sludge formation and acid formation.

2. Oils which deteriorate with respect to sludge formation but which still retain resistance to acid formation and corrosiveness.

3. Oils which deteriorate with respect to acid formation and bearing corrosion but which still resist oxidation of the sludge-forming type.

4. Oils which are highly stable and resist all forms of oxidation even under severe oxidizing conditions.

■ Underwood Tests – Conclusions

1. The agreement among the laboratories which ran cooperative Underwood tests is surprisingly good considering the many variations in details of test procedure. With more nearly uniform test conditions and procedure it would appear that agreement among laboratories would be considerably better and would quite conceivably reach a very satisfactory state of reproducibility.

2. The Underwood apparatus, when run for 10 hr at 325 F with 0.01% Fe_2O_3 (as naphthenate), correlates quite satisfactorily with 36-hr Chevrolet engine tests on five of the six reference oils; the single exception being the corrosion rating of oil C-3. This discrepancy may be considered of relatively minor importance since both the Underwood and the Chevrolet engine rate oils of the C-3 type as having limited oxidation resistance both as to sludge formation and acid formation. The essential difference lies in the fact that the oxidation acids formed are non-corrosive to sensitive bearings in the engine but are corrosive to bearings in the bench test.

3. Underwood test conditions which appear to produce the best correlation with Chevrolet engine tests are outlined in Table 6. These conditions represent the approximate average of those used by the laboratories reporting the most consistent results.

Table 6 – Average Underwood Test Procedure

325 F	10 hr
1500-c.c. Oil Sample	
0.01% Fe_2O_3 as Naphthenate	
2 x 10 in. Copper Baffle	
1 Cadmium Bearing	
10 psi Oil Pressure	
Optional—	
1 Copper-Lead Bearing	
5-hr Test Data	
Cleaning Method	

4. It is generally agreed that the Underwood apparatus, or any other bench test apparatus, cannot be used to predict service performance without additional substantiating tests in engines. However, the Underwood apparatus is indicated to be a very useful sorting device to differentiate the stability and corrosion properties of poor, mediocre, and good oils, particularly as related to research and development on improved oils and additive compounds.

■ Lauson Engine Tests

Table 7 is a summary of the Lauson engine operating conditions used by the cooperating laboratories.

Although there are a wide variety of Lauson test procedures in use, with no two laboratories reporting exactly the same set of operating conditions, it will be noted there are two general types of operating procedures:

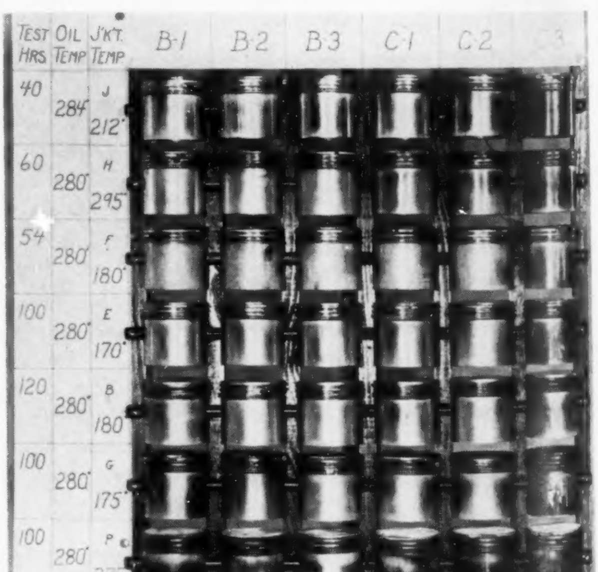
A. The oil stability or varnish procedure, involving high crankcase oil temperatures in the order of 280 F and moderate coolant temperatures in the order of 180 F.

B. The detergency or ring-sticking type of procedure, involving moderate crankcase oil temperatures in the order of 225 F and very high coolant temperatures in the range from 340 to 400 F.

Table 7 – Lauson Test Conditions

Speed		Test Time	
No. of Laboratories	rpm	No. of Laboratories	hr
1	1200	3	25
8	1600	1	30
3	1700	3	40
6	1800	1	48
Load			
No. of Laboratories	hp	1	54
4	3 (Full)	2	60
1	2½	5	100
12	1.3 to 1.6	2	120
1	1.0	1	144

	Temperatures	
No. of Laboratories	Oil, F	Jacket, F
5	280	170 to 180
1	280	212
1	280	275
1	280	295
1	320	212
3	225	340 to 350
1	240	345
1	212	320
3	225 to 230	400
1	300	375
1	220	300

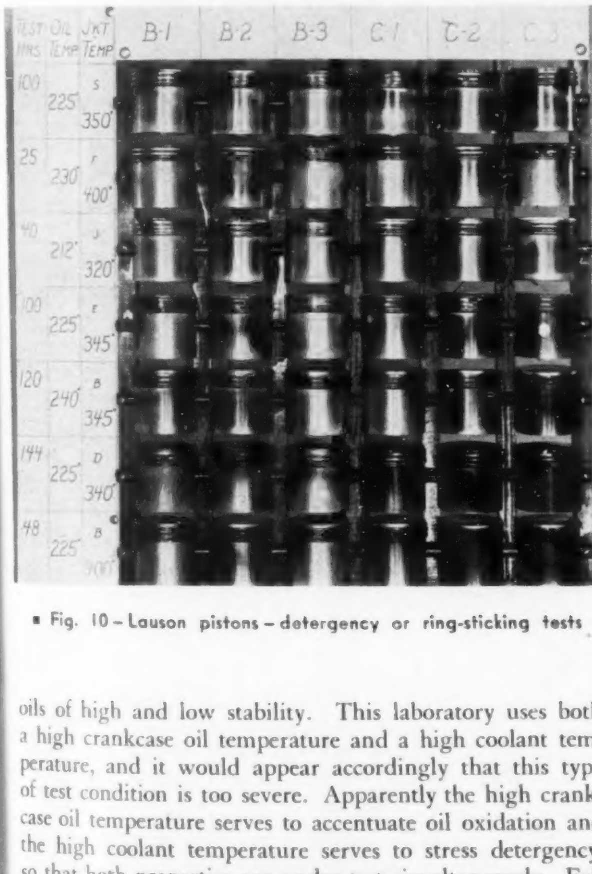


■ Fig. 9 – Lauson pistons – varnish or oil stability tests

■ Lauson Oil Stability Tests

Fig. 9 shows seven sets of Lauson pistons from varnish or oil-stability type test procedures. In spite of the wide variety in specific test conditions, there is very good general agreement on the piston varnish ratings. All seven laboratories rate oil C-3 as a heavy varnish former, the other five reference oils being rated as substantially varnish-resistant. This is in agreement with the 36-hr Chevrolet tests. Laboratories B, G, and P rate oil B-1 as showing appreciable piston varnish. This is also in agreement with the Chevrolet tests. Oil B-1 was rated as satisfactorily stable in the Chevrolet tests at 36 hr but did develop appreciable varnish and sludge after 67 test hr. The oil can therefore be considered as stable or somewhat unstable depending upon the severity of the stability test conditions. The Lauson engines show this same differentiation, producing either clean pistons or somewhat varnished pistons depending on the severity of the test conditions.

Laboratory P shows all oils as developing appreciable varnish, and indicates a rather poor differentiation between

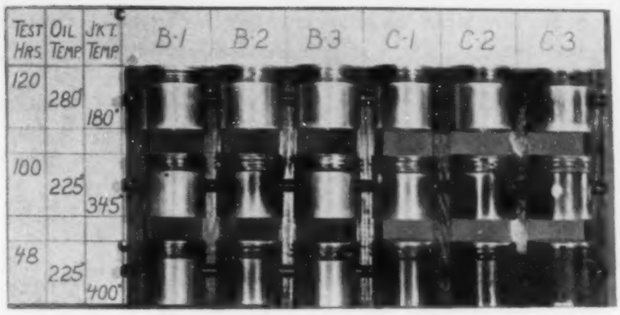


■ Fig. 10 – Lauson pistons – detergency or ring-sticking tests

oils of high and low stability. This laboratory uses both a high crankcase oil temperature and a high coolant temperature, and it would appear accordingly that this type of test condition is too severe. Apparently the high crankcase oil temperature serves to accentuate oil oxidation and the high coolant temperature serves to stress detergency, so that both properties are under test simultaneously. Following the practice of most laboratories, it appears desirable to make a definite distinction between stability tests and detergency tests.

■ Lauson Detergency Tests

Fig. 10 shows seven sets of Lauson pistons from detergency or ring-sticking procedures. Again there is good general agreement among the laboratories in spite of the wide variation in test procedure details. Oils B-1 and



■ Fig. 11 – Lauson pistons – three sets of test conditions

B-3 are rated best, showing the minimum of skirt varnish formation and ring sticking. The other four oils show relatively limited detergency properties with C-3 rated as worst. These tests appear to correlate satisfactorily with the detergency information available on the six reference oils.

Laboratories F and J tend to rate the oils too close together, with only a limited spread between the best and the poorest. In the case of laboratory F, the test time of only 25 hr may account for this condition. With laboratory J, the short time of 40 hr coupled with the relatively low coolant temperature of 320 F may account for the somewhat poor differentiation.

Of interest is the similarity of test results shown by laboratories D and B (see the bottom two rows of pistons in Fig. 10). Laboratory D runs for 144 hr with a jacket temperature of 340 F, whereas laboratory B runs for only 48 hr but with a jacket temperature of 400 F. Where an accelerated detergency test is desired, it appears that use of a very high coolant temperature will produce piston varnish conditions substantially the same as a somewhat lower coolant temperature for longer test time.

Some of the laboratories did not send in the pistons from their Lauson runs for inclusion in the group photographs. However, individual photographs as submitted by these laboratories show the same general conditions of good agreement as illustrated by Figs. 9 and 10.

Fig. 11 shows three sets of Lauson pistons from three typical sets of test conditions. Under the stability or varnish type of procedure, utilizing a 280 F oil temperature and 180 F jacket temperature, oil C-3 is rated as a heavy varnish former, and also shows considerable ring fouling with sludgey residues, whereas the other five oils left clean pistons. With this type of operating condition, there was no evidence of ring sticking with any of the oils, and the varnish deposits, where formed, were of a sticky or resinous nature.

With the detergency-type procedure, utilizing an oil temperature of 225 F and a coolant temperature of 345 F, oils B-1 and B-3 are rated good, oil C-1 as second best, and the other three reference oils as poor, with C-3 rated as worst. Under this type of operating conditions, piston-skirt varnish formation was heavy with the oils of limited detergency, as was ring fouling with coky residues. Varnish from these test conditions was hard and glossy. However, very little ring sticking was developed even with the poorest oils.

With the accelerated 48-hr detergency test using a 400 F coolant temperature, the piston conditions with respect to

varnish formation are remarkably similar to the 100-hr test procedure using the 345 F jacket temperature. However, with the 400 F jacket temperature, oils B-2, C-2, and C-3 show marked ring sticking. It would appear, therefore, that a distinction must be made between engine tests to evaluate detergency and tests to evaluate ring sticking. With the Lauson engine, a jacket temperature in the order of 400 F is apparently necessary to develop ring-sticking tendencies.

Fig. 11 also serves to illustrate the differences between oil stability and oil detergency, and the effects of engine operating temperatures in evaluating these properties. High crankcase oil temperatures serve to emphasize the stability and oxidation resistance properties of oils, whereas high piston or coolant temperatures serve to emphasize detergency characteristics, and still higher piston temperatures introduce ring-sticking tendencies.

The six reference oils include the four basic varieties of engine lubricating oils:

1. Those which have both limited stability and limited detergency.

2. Those which are limited as to detergency but which are stable and oxidation resistant.

3. Those which are limited as to stability but which possess detergency.

4. Those which possess both high stability and high detergency.

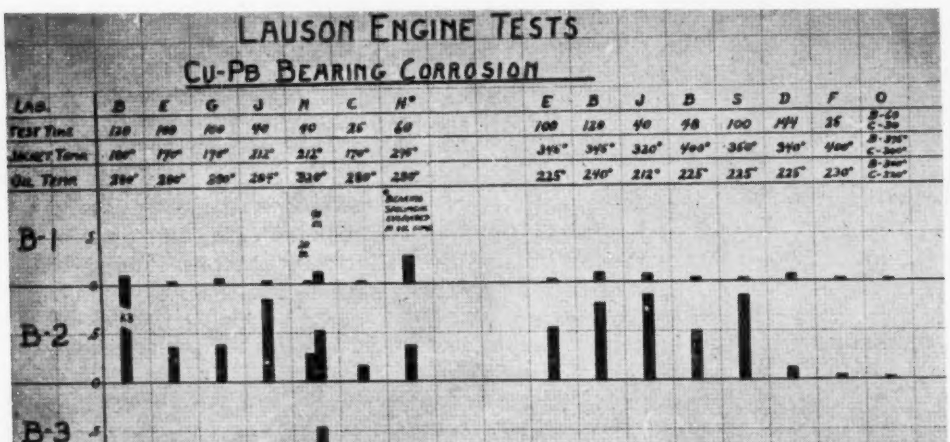
Selection of the proper Lauson operating conditions, particularly as to oil and coolant temperatures, apparently rates the oils in their proper order, when separate tests are run to evaluate each property. When test conditions are set up to attempt evaluation of both stability and detergency simultaneously, misleading results are apt to be secured.

Lauson Bearing Corrosion Data

Fig. 12 shows copper-lead bearing corrosion data reported by the cooperating laboratories. Considering the wide variation in test conditions, the agreement among laboratories is quite good and the oils in the majority of the tests are rated in the same order by the Lauson engine as the Chevrolet engine.

The data reported by laboratory N are of particular interest. This laboratory used a 320 F crankcase oil temperature, which is considerably higher than used by the other laboratories. The Lauson bearing corrosion data after 20-hr test periods check with the Chevrolet ratings, whereas the corrosion data after 40-hr periods tend to rate oil B-3 as too corrosive in comparison with oils B-1 and B-2. It is generally recognized that bench test conditions of such extreme severity can be set up, that any oil will deteriorate very rapidly, irrespective of its service performance record. It appears that a 320 F oil temperature in the Lauson engine approaches this category; and if oil temperatures in this range are used, the oxidizing conditions are so accelerated that very short test times are necessary.

Laboratory H suspended copper-lead bearing segments in the oil sump rather than inserting them in the connecting rod. It will be noted that the order of rating of oils



■ Fig. 12 – Lauson engine tests – copper-lead corrosion data

B-1, B-2, and B-3 is quite poor; and it appears, therefore, that accurate bearing corrosion data can only be secured by running the copper-lead bearings in the connecting rod and not as test specimens in the oil sump. Laboratories C and F show only a small spread in bearing corrosion among all six reference oils. This is apparently due to the short time of 25 test hr used by these two laboratories; and it appears, in order to secure accurate copper-lead corrosion data in the Lauson engines, that test times in the order of 100 hr are desirable.

Laboratory D also reports only a small spread among the six reference oils on copper-lead bearing corrosion, there being but little difference between the most corrosive and the least corrosive oils. This laboratory uses a special connecting rod, crankshaft, and other equipment in their Lauson engines. Apparently the drastic design changes made in the Lauson by this laboratory have a marked effect on the bearing operating conditions, and the corrosion results are thrown considerably out of line compared to the majority of cooperating laboratories.

Laboratory O reports poor differentiation in bearing corrosion ratings, and shows oil B-3 to be more corrosive than B-2, which is contrary to practically all of the other corrosion evidence. The 300 F crankcase oil temperature used in testing the "B" series of oils apparently produces questionable results similar to those of laboratory N as discussed above. The quite moderate test conditions of 300 F coolant temperature and 220 F oil temperature used with the "C" series of oils do not seem to be sufficiently severe to distinguish stability characteristics adequately.

Many of the cooperating laboratories reported only partial oil analysis data on the drain samples from their Lauson runs and there were accordingly not sufficient oil deterioration data to warrant tabulation. The oil drain inspection data, where reported, varied widely depending on the test conditions, particularly with respect to crankcase oil temperature and test time. However, what oil data were reported showed the same general trends indicated by the pistons and bearing corrosion data. With operation of the Lauson engines under more uniform test conditions, oil deterioration could most probably be brought into as good agreement as has been developed with the standardized Chevrolet tests.

■ Lauson Engine Tests – Conclusions

1. The Lauson engine is a remarkably versatile oil testing device, and with proper selection of operating conditions can apparently be made to rate oils as to their stability, bearing corrosion, detergency, and ring-sticking properties in line with full-scale engine tests.

Test time, oil temperature, and coolant temperature are the most important variables. Engine speed does not seem to be critical over a considerable range. Load also does not seem to be critical as to oil evaluation, although the laboratories reporting use of loads in the order of 1.5 hp seem to have little trouble with mechanical failures of engine parts, whereas the laboratories operating the Lausons at full load report difficulties with connecting-rod and crankshaft breakage and in some instances have resorted to quite extensive changes in the engines to overcome such troubles.

2. The cooperative Lauson test data serve to illustrate the importance of operating temperatures on oil evaluation. High crankcase oil temperatures place emphasis on oil stability or oxidation resistance, while high coolant or piston temperatures place emphasis on detergency.

It is interesting to note that a considerable number of laboratories working independently have developed Lauson test procedures which, while varying considerably as to detail, agree surprisingly well as to basic oil evaluation and as to differentiation between the properties of stability and detergency.

3. Table 8 is a set of Lauson operating conditions for evaluating oil stability and bearing corrosion tendencies.

Table 8 – Lauson Test Procedure for Oil Stability and Bearing Corrosion

Oil Temperature	280 F
Coolant Temperature	180 to 200 F
Test Time	100 to 120 hr
Engine Speed	1600 to 1800 rpm
Load	1.3 to 1.6 hp

Table 9 is a set of conditions for evaluating detergency and bearing corrosion, and Table 10 a set of conditions for an accelerated detergency and ring-sticking procedure. These

Table 9 – Lauson Test Procedure for Detergency and Bearing Corrosion

Oil Temperature	225 F
Coolant Temperature	345 to 350 F
Test Time	100 to 120 hr
Engine Speed	1600 to 1800 rpm
Load	1.3 to 1.6 hp

Table 10 – Accelerated Lauson Test Procedure for Ring Sticking and Detergency

Oil Temperature	225 F
Coolant Temperature	400 F
Test Time	24 to 48 hr
Engine Speed	1600 to 1800 rpm
Load	1.3 to 1.6 hp

respective sets of test conditions represent the general averages of the various procedures used by the cooperating laboratories which reported the most consistent results on the six reference oils.

4. It is possible to set up test conditions which will rate all oils as poor, regardless of their service performance records. Apparently in the Lauson engines such conditions are approached by use of crankcase oil temperatures in excess of about 280 F or by use of a combination of both very high oil and coolant temperatures simultaneously.

As with other types of bench tests, Lauson engine data cannot be used to predict, unfailingly, service performance unless further substantiated by tests in full-scale engines. However, the Lauson engine is indicated to be a most valuable instrument for differentiating between oils either as to stability and corrosion properties or as to detergency characteristics.

■ Other Bench Test Data

A number of the cooperating laboratories reported test data on the six reference oils from other types of test apparatus. While several of these test methods have been described in technical publications they have not been used as widely as the Underwoods and Lauson engines.

Comparison of tests on the six reference oils by these various apparatus with the Underwood and Lauson data is of considerable interest.

Indiana Stirring Oxidation Tests

48 hr at 330 F

Oil No.	Neutralization No.	Sludge %	Seconds Viscosity Increase (100 F)	Varnish 10=None 1=Heavy
B-1	3.3	2.07	588	10
B-2	9.3	4.04	Too Viscous	10
B-3	6.4	2.36	255	10
C-1	4.9	2.94	279	9½
C-2	1.3	0.15	87	9
C-3	13.7	7.58	Solid	−1

The Stirring Oxidation Test appears to rate oils B-3 and C-1 as quite poor in stability, sludge, and acid formation compared to the Chevrolet, Underwood, and Lauson tests. Conversely, the Stirring apparatus appears to rate oil C-2 as having better stability than do the Chevrolets, Underwoods, and Lausons.

LABORATORY L

MacCoull Corrosion Tests

10 hr at 350 F

Oil No.	Bearing Corrosion Loss, mg		Neutralization No.	Undissolved Sludge, %
	4 hr	10 hr	10 hr	10 hr
B-1	10	115	3.8	0.48
B-2	116	267	14.5	0.31
B-3	9	89	2.6	0.15
C-1	11	15	0.6	0.01
C-2	51	222	11.5	0.85
C-3	66	252	11.3	1.92

The MacCoull Corrosion Tests on the six reference oils appear quite similar in overall ratings to the Underwood tests. It will be noted that the MacCoull tester rates oil C-3 as highly corrosive, the same as does the Underwood, although C-3 oil was non-corrosive in the Chevrolet tests.

LABORATORY O

Strip Corrosion Tests

72 hr at 300 F

Oil No.	Naphtha Insoluble, % 72 hr	Copper-Lead Corrosion 48 hr	Copper-Lead Corrosion 72 hr
B-1	0.56	0.003	0.006
B-2	0.01	0.243	0.423
B-3	0.01	0.040	0.125
C-1	0.19	0.003	0.003
C-2	0.03	0.021	0.143
C-3	2.65	0.103	0.194

Strip Corrosion Tests

72 hr at 325 F

Oil No.	Naphtha Insoluble, % 72 hr	Copper-Lead Corrosion 48 hr	Copper-Lead Corrosion 72 hr
B-1	1.84	0.004	0.004
B-2	0.01	0.024	0.054
B-3	1.71	0.014	0.022
C-1	1.21	0.003	0.003
C-2	0.82	0.001	0.005
C-3	3.65	0.005	0.009

The Strip Corrosive Tests, reported by laboratory O, are apparently a modification of the original Indiana Oxidation Test. The Strip Tests at 300 F rate oil B-3 as having about the same degree of corrosiveness as oil C-2, which is contrary to the Chevrolet and the other bench tests. The sludge formation ratings are in the same order as the Chevrolet engine sludge ratings, but the bearing corrosion correlation seems poor.

The Strip Corrosion Tests at 325 F rate all six of the reference oils very close together on corrosion, and appear to show inadequate spread compared to the Chevrolet tests and the other bench tests. The naphtha insoluble values at 325 F also do not appear to correlate as well as the same type of tests at 300 F.

Sohio Oxidation Tests

Time – 36 hr
Temperature – 280 F for "B" Oils
265 F for "C" Oils

Oil No.	Varnish Rating %	Copper-Lead Bearing Corrosion mg/10cm²	Viscosity Increase (100 F)	Sludge %
B-1	70	46	410	0.85
B-2	93	72	1050	Trace
B-3	95	31	16	Trace
C-1	97	12	150	0.15
C-2	99	27	580	Trace
C-3	87	32	1160	2.4

The Sohio Oxidation Tests produce varnish ratings substantially in line with the Chevrolet ratings with the possible exception of oil C-3, which may be rated somewhat too high by the Sohio Test. The copper-lead bearing corrosion ratings from the Sohio Test show quite limited spread between the six reference oils, with the result that the corrosion ratings of oils B-3, C-2, and C-3, in particular, are not differentiated as well as they might be. The oil deterioration values of viscosity increase and sludge formation fall in line with Chevrolet ratings.

LABORATORY R

Series 30 Ethyl Engine Tests

Speed – 1200 rpm | Load – 2 hp
Oil Temperature – 300 F | Jacket Temperature – 212 F

Oil No.	Test, hr	Piston-Skirt Deposit Rating, %	Ring Sticking Rating	Bearing Corrosion
B-1	70	50	100% Free	0.010
B-2	60	50	100% Free	0.030
B-3	100	85	100% Free	0.020
C-1	70	50	100% Free	None
C-2	50	50	100% Free	0.030
C-3	30	0	100% Free	0.020

The series 30 Ethyl engine, operated under conditions apparently intended to evaluate oil stability, produces results as to piston varnish deposits very similar to Lauson engines when run under stability test conditions. Photographs of the pistons from the Series 30 Ethyl Engine Tests were substantially identical to the Lauson pistons shown on Fig. 9.

Bearing corrosion data from the series 30 Ethyl Engine tests show essentially no difference between the six reference oils. It is believed, however, that babbitt bearings are used in the series 30 engine so that corrosion tendencies are not intended to be evaluated.

Requirements for Carburetor Air Filters for Aircraft Engines

continued from page 37

turbocompressor after the intercoolers. This location was selected as the point that will produce the least effect on the performance of the supercharger. The filter is located after the intercoolers for two reasons: First, so that heat control can be accomplished with the intercooler shutters; and secondly, so that the normal flow of air through the viscous impingement element will be at a reasonable temperature.

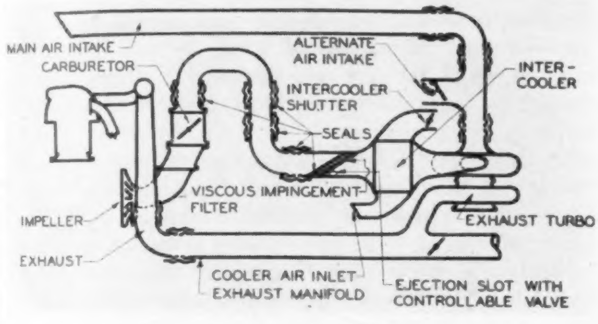


Fig. 12 – Installation diagram of an induction system with an exhaust turbosupercharger and a viscous impingement filter

In a turbo-installation there is some advantage at high altitude in being able to stop bleeding air out of the induction system on the pressure side of the turbosupercharger, and for this reason the diagram indicates controllable flaps at the ejection slots.

In conclusion, it is pointed out that the diagrammatic installations that have been shown are believed to give the engine the protection it needs against damage by sand, with the least complication and least added weight to the airplane, without upsetting the carburetor metering, and without imposing any responsibility or added duties on the pilot.

New Methods for the Evaluation and Recording of Piston Skirt Deposits

continued from page 44

of reflectivity. This term was introduced as a result of a study of photographic emulsion characteristics by Hurter and Driffield. In their work it was found convenient to express the opacity of the negative as a logarithm which was called the density. The concept of density has been carried over to work on the reflectivity of photograph papers and is now used as the logarithm of the reciprocal of the reflecting power.

The reflecting power of an object is measured under conditions of 45-deg illumination to eliminate surface effects and to correspond more nearly with the visual response of the observer viewing the object under indirect light.

Piston lacquer deposits are most conveniently rated in terms of reciprocal reflecting power instead of densities because of the expanding scale obtained for the lighter deposits.

APPENDIX II

Photographic Technique

After initial speed of exposure is determined, some variation from piston to piston is necessary, depending upon the amount of deposit on the piston to be photographed. Changes in time are effected by a change in the diaphragm opening of the lens. In determining the exposure for Type B Kodachrome, correcting color filters are used in order to secure an effect of 3200° K on the films. Tests may be made with direct positive paper for highlight detail. Interchangeable light sources are used. A tungsten photoflood lamp, R2, is used for all Kodachrome photography, and also for the black-and-white photography of small pistons. Large pistons are photographed by fluorescent light in order to obtain an even illumination throughout their length; Kodachromes by fluorescent white light are not true in color, being too green (even with the most suitable correction filters), but have been adequate for comparison purposes. When the demand for truer colors on these larger pistons occurs, a tungsten light source can be devised.

The slit is covered over, leaving open only an area as large as the piston to be photographed. If a lens of suitable focal length is not available, a plus or minus diopter supplementary lens will usually give a combination that is workable.

Cooperative Research Comes of Age

continued from page 51

made on the preparation of vehicles for storage and reconditioning for service, in so far as fuels and lubricants are concerned.

Those of us familiar with small-boat operation appreciate keenly the difference in arduousness and risks as between fair and foul weather. In fair weather, the course is clearly discernible and easily followed. The skipper and crew have easy jobs, with plenty of time for relaxation. Foul weather may come in the shape of high winds, which toss about both buoys and boat and make the task of finding and following the course difficult. It sometimes takes the form of fog, which may totally blank out your surroundings and lure the mariner caught out without a

compass into sailing a circular course. Even a lighter fog is baffling, since it distorts the shapes of familiar landmarks. Or the shifting gray of a driving rain may alternately reveal and obscure channel markers. Under such circumstances both captain and crew must be on deck, continuously alert. Every eye is strained to search out the course, every set of muscles is ready for emergent and strenuous action. Full compensation is the thrill of bringing your craft into a safe harbor, with gear intact, having successfully pitted your skill and strength against unpredictable and uncontrollable forces.

Launched by the petroleum and automotive industries, the Cooperative Research Council has slid down the ways into the troubled seas of wartime industry. It has, and will receive, the needed service of its captains and crews, participants in laboratories of both industries, in untiring zeal, technical skill, and harmonious cooperation. Our immediate goal is to help bring to our nation a day of relief and exultation such as it knew on November 11, 1918.

For the post-war future we envisage ever-extending voyages of investigation. Our allies, the Chinese, the most rational people in the world, have no word in their language for either yes or no. So it is with research, which may never reach a static state of definite answers to every question; since it is a dynamic force, adapting itself to ever-changing current needs and finding each solution only a bend in the river opening up further vistas of inquiry. In a post-war world our type of research will have both economic and social appeal. It deals with the effective utilization of our natural resources, now subjected to the drain and wastage of war. Substituting coordination for competition as between our two great industries, it will be in harmony with what we hope will be the new way of life – achievement through voluntary peaceful cooperation.

Control of Oil Consumption in the High-Speed Four-Cycle Automotive Diesel Engine

PROBABLY the greatest single factor in successful control of oil consumption is the cylinder barrel. When it remains a true cylinder little difficulty is experienced, but it becomes a real source of trouble, remaining forever a disturbing or destroying factor, should it be subject to fixed or variable distortion.

Under the classification of "fixed" distortion of the bore will normally be found the physical forces traceable to stud loading, insufficient deck rigidity or faulty design relation among stud bosses, cylinder bore, and cylinder deck. It should hardly be necessary to mention also that a cylinder barrel should be physically capable, in the original design, of holding itself by its own strength, in addition to satisfying all other design requirements. After exercising all the generosity permitted by economy (of space particularly) the cautious designer should at this point offer foundry people the opportunity to suggest coring or modifications that will, in a proposed design, allow them to bring forth castings of uniform section which will retain the intended design features.

Under "variable" distortion of the barrel, the common disturbing possibilities are the cooling system and the combustion chamber. It would seem that many years of development should have taught us nearly all there is to know about the application of desirable cooling principles within a cylinder block. However, each new block or head design includes changes of water space and passages sufficient to require restudy of water flow and heat transfer. It is not safe to assume that water flow follows the design pattern, and the system should be checked by the tedious process of thermocouple investigation. Timely investigation offers us the only opportunity to discover and correct undesirable temperature differences around the bores. Regardless of the apparent excellence of a directed water flow, we would probably be quite surprised if we could see the actual behavior of the water within the jackets of a multi-cylinder engine. The value of cooling-water control is not questioned, but our knowledge of it to date is such that we should not attempt fancy directional control of cylinder-bore cooling without considerable experimental study. In fact we should carefully avoid directional flow toward the bores, making sure only that the greatest possible area is exposed to water cooling, and that the cooled areas are simple symmetrical shapes.

Regarding the combustion chamber, or system, it may be said that the automotive diesel engine presents a more severe problem than the gasoline engine of equal size, output, and speed. The present state of the art permits a choice (not sharply defined) between two basic sets of conditions. One may select the open combustion chamber or a variation thereof, accepting the severe load upon rings, since combustion control is largely a matter of injection and timing. On the other hand, the designer may select a combustion chamber of the air-cell or precombustion chamber type in any of its numerous variations. While the combustion directly above the piston might be softer and at a lower rate of pressure rise in the precombustion or air-cell type, such systems are not always uniform in temperature or pressure at all points radially located around the piston or cylinder. The obvious result can be cylinder distortion of the variable type affecting the top end of the bore proper, or this section in combination with the cylinder deck.

Note that all of the above relates to the path followed by oil that is usually referred to as "consumed." If a fair degree of success has been achieved in physical design, and combustion is well controlled, an engine may still fail to respond properly to treatment that is usually effective. Within the enclosed crankcase of the common breed of four-cycle diesel engine are found the usual elements employed to transfer combustion energy to rotative force at the shaft. Essentially these parts are similar in arrangement and mounting to those of the gasoline engine, and of course play a part in oil control. The designer faces the usual consideration of gear- and accessory-drive oil feed, returns from oil pump bypass, oil cooler, and filter, in addition to possible excessive quantities of oil that may be thrown into the cylinder bores by interference between moving parts and oil streams.

Excerpts from the paper of the same title by A. T. Stahl, Mack Mfg. Corp. – International Plainfield Motor Co., presented at the ASME 15th National Oil and Gas Power Conference, the SAE Diesel-Engine Activity cooperating, Peoria, Ill., June 18, 1942.

A BRIEF SURVEY of the PRINCIPLES of PRESSURE WATER COOLING

by JAMES E. ELLOR

Chief Research and Development Engineer, Rolls-Royce, Ltd., Derby, England

As speeds and operational altitudes of modern aircraft continue to increase, it is becoming more and more important that the total drag of the airplane be reduced while the rate of heat dissipation per unit frontal area of radiator be kept as high as possible.

The standard method of increasing the temperature difference between cooling medium and coolant has been to use ethylene glycol as a coolant, because its boiling point is much higher than that of water; however, in its pure state glycol has various disadvantages that are not present when a pressure water system is used.

This is a sealed system for making use of the physical characteristics of the increase in boiling temperature with pressure. When the radiator receives more heat from the engine than it is dissipating, a small quantity of steam is generated inside the cylinder jackets. The resulting increase in pressure will cause the temperature to rise until a balance is restored between heat rejection and radiator dissipation.

In discussing the results of his experiences with pressure cooling, Mr. Ellor has included design details of a header tank and a suitable thermostatic header tank relief valve.

■ ■ ■

WITH the increase in engine power and speed of modern aircraft, a great deal of thought has been given to the rate of heat dissipation per unit frontal area of radiators. In order to reduce the total drag of the airplane, two factors directly influencing radiator heat dissipation per unit of matrix area must be considered. They are mass flow of air and temperature difference between cooling medium and coolant. Increasing the temperature difference between the two media furnishes a marked improvement without affecting the drag.

The obvious course to adopt is to use a coolant having a higher boiling point. For some time past it has been standard practice to use ethylene glycol, which has a boiling point of 387 F at sea level pressure. In its pure state, there are, however, distinct disadvantages with this coolant, a few of which are itemized as follows:

1. The rate of heat transfer from the cylinder walls to the cooling medium is less than with water. The effect on the engine is to raise the metal temperature approximately 90 F, and this limits the maximum output of the engine. A good illustration of the reduced rate of heat transfer is furnished from heat-flow tests on radiators. In a given size radiator where the air flow and temperature are constant, the heat dissipation per 180 F temperature difference is 89% with 30% glycol, 70% water; and 66% using 97% pure glycol. These figures are relative to 100% for water.

2. Since ethylene glycol is hygroscopic, care must be taken to see that the water content does not exceed 5%. The boiling point of 95% glycol at sea level pressure is 325 F, falling to 248 F as the pressure drop corresponds to an altitude of 27,000 ft. Thus at altitudes of 30,000 ft and over, the outlet temperature of the coolant will fall below 248 F unless some means of pressurizing the glycol is resorted to.

3. Its corrosive action on metals, particularly aluminum, necessitates the introduction of an inhibitor.

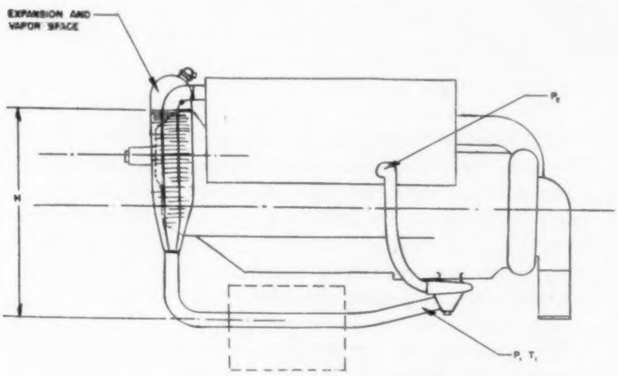
■ Pressure Water System

This is a sealed system and use is made of the physical characteristics of the increase in boiling temperature with pressure. When the rate of heat rejection exceeds the rate of heat dissipation of the radiator, a small quantity of steam is generated inside the cylinder jackets. Any formation of steam will tend to increase the total volume of the system, but since this is constant the temperature rises until the balance is restored between heat rejection and radiator dissipation for the system to operate satisfactorily during the boiling suppression periods. A vapor space must be provided in the system at or above the level of the cylinder outlet from the engine. It is, of course, necessary to provide an air space in the header tank for separation and expansion, and this will also serve the same purpose. Thus it will be seen that water outlet temperatures far in excess of those required can be obtained, the limit being imposed by permissible pressures. A figure of 25 to 30 psi above the surrounding atmospheric pressure is advocated, which permits an outlet temperature of 257 to 263 F at 35,000 ft when using 30/70 glycol-water mixtures. When pressure is generated in the system to maintain the balance between heat rejection and dissipation, the conditions in the header

[This paper was presented at the National Aircraft Production Meeting of the SAE, Los Angeles, Calif., Oct. 1, 1942.]

tank approximate to boiling point, and flow characteristics into the pump can become critical unless certain factors are observed.

To illustrate this, consider a simple system with the radiator removed and where an open-top header tank is used whose height above the pump inlet can be adjusted, the engine outlet flowing freely into the tank. At the boiling temperature, there will be no flow into the pump when the header tank is at the pump level. Raising the tank level, however, will produce a flow proportional to the square root of the head above the pump inlet even though the header tank and the pump inlet are both at the boiling temperature. If, in addition, the bulk of the coolant is permitted to run in direct circulation using the header tank water as make-up, the head will be further increased proportional to its kinetic energy. This effect is shown graphically on Fig. 1, which indicates diagrammatically a direct-flow pressure cooling system where the tank is located at the highest point above the pump inlet around

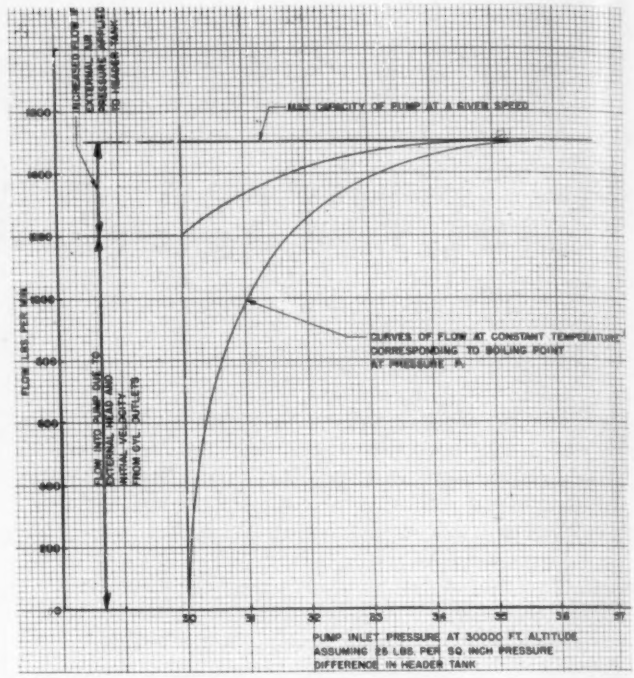


■ Fig. 1 – Direct-flow system

the nose of the engine. The vapor space is indicated and small holes are provided on the inside of the water outlet to allow for steam separation.

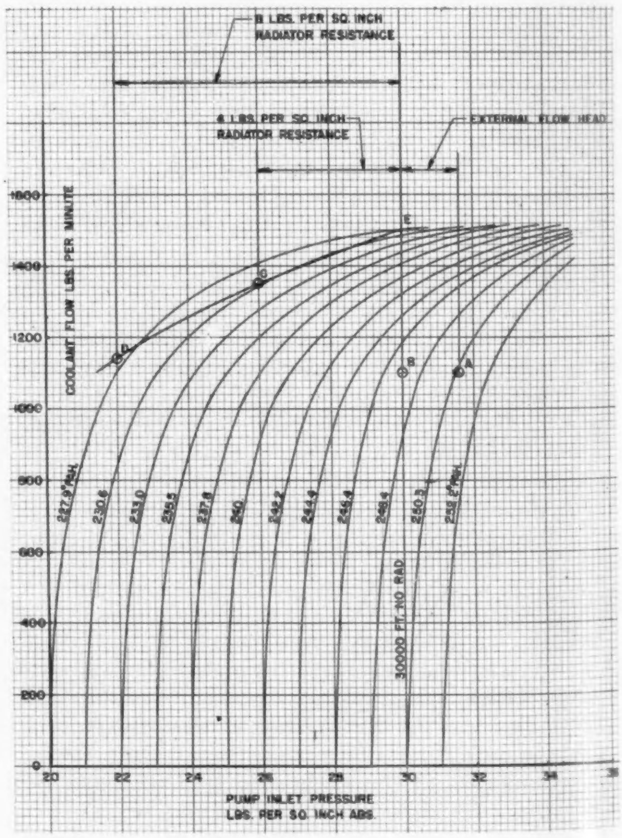
Incidentally, if the header tank is placed at the rear, it must be sufficiently high to cover the cylinder heads in the tail-down position on the ground. This usually entails the raising of the cowl line and excludes the desirable feature of standardizing header tanks on a given engine. Fig. 2 shows curves at boiling temperatures which would be obtained at the pump inlet at 30,000 ft, assuming a pressure difference of 25 psi in the header tank. On this particular pump operating under boiling conditions, the flow is raised from 0 to 1200 lb per min by locating the tank in the position shown and making use of the kinetic energy from the outlet. Airplanes are in operation using different systems where the radiator is located on the pressure side of the pump; an increased flow up to the maximum capacity of the pump can be obtained by the application of pressure to the header tank as indicated in Fig. 2. Locating the radiator in the inlet side of the pump system will produce the same flow as in the previous example, if the increased resistance was compensated by a drop in temperature equal to the difference of the corresponding boiling points at those pressures. If, however, the drop in temperature exceeds that which corresponds to the drop in pressure, the pump depression will increase, producing a corresponding increase in flow.

Referring to Fig. 3, a series of curves are drawn for a



■ Fig. 2 – Curves of flow at constant temperature corresponding to boiling point at pressure P_1

particular pump of the flows at various inlet pressures and boiling temperatures. At 30,000-ft altitude with a pressure difference in the header tank of 25 psi, boiling conditions



■ Fig. 3 – Pump characteristics under pressure showing effect on flow of different radiator resistances – assuming water as coolant and 25 psi pressure above surrounding atmosphere

occur at 250 F using water. With no external head on the pump, there will be no flow. With the external head and kinetic energy shown in the set-up in Fig. 1, a flow indicated in point B (Fig. 3) is obtained at the same temperature of 250 F. Inserting the radiator, which we will assume has 4 psi additional resistance, increases the flow from point B to point C. This point is located by assuming an engine develops 1200 hp at 30,000 ft with the particular flow characteristics of the pump shown on the curves. The drop in temperature of the radiator is obtained by dividing the heat rejected by the weight of water. Where the curve at this particular temperature intersects the 4-psi pressure difference line indicates the flow. Making a similar calculation for an 8-lb radiator resistance, point D is located, which is lower than point C. Point E is a hypothetical point which indicates a radiator with no resistance. From these curves it will be seen that the quantity of circulation is not critical, assuming a reasonable initial head, and providing the pump characteristics are suitable for operating in the vicinity of boiling temperatures. Making use of the external head and momentum is very necessary, particularly at low engine speeds. To operate this system successfully, an efficient means of separating the steam must be provided and air must be prevented from entering the pump feed line.

■ Header Tank Design for Pressure Cooling

The design and arrangement of the header tank is one of the most important parts of a successful system. In addition to supplying make-up water and providing the external head, it must also be capable of efficiently separating any steam, air, or vapor and prevent any gaseous substance from penetrating into the feed line. It is also in the air-space portion of the tank that the pressure control is operative. The layout of one very successful type of tank is shown in Fig. 4.

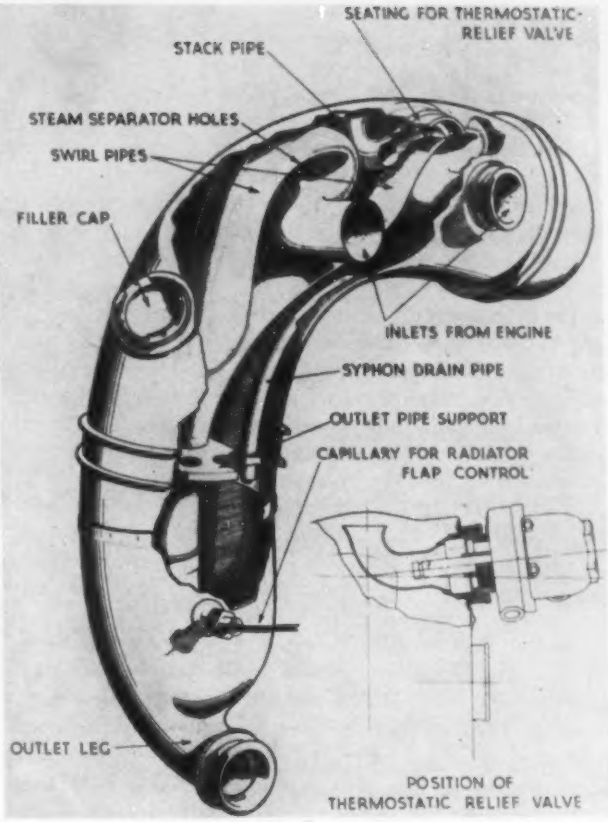
When the engine is run cold, the air space in the cylinder jacket is filled with coolant drawn from the header tank, and consequently the level in the tank will fall to the starting level. Under these conditions, there should be a minimum quantity of coolant remaining in the tank. As the system is heated, a displacement or expansion of coolant occurs from the engine cylinder jacket, and consequently the level in the header tank rises to the normal running level. Above this level, a residual air space is required for the following reasons:

1. Separation.
2. The control of air and vapor pressure.
3. To prevent loss when the relief valve opens.

The separation for pressure-cooled installations is obtained by means of two pipes in the shape of scrolls, through which the coolant passes immediately after entering the header tank. On the inner diameter of the scrolls, separation holes are placed so that the separated air and vapor is discharged into the header tank air space. Thus, in order that the separation shall be efficient, it is necessary for the separation holes to be situated above the running level. This requirement will obviously depend on the shape of the header tank and the scroll pipes; this, in general, is found to equal an air space of 6 to 8 pt, and to give a level of 3 to 4 in. below the venting point, thus fulfilling the other conditions.

The pipes or scrolls which form the separators are, in effect, part of a closed circuit, so that the coolant entering the tank passes round the separator, and then direct to the outlet without passing into the body of the tank, thus conserving the kinetic energy of the coolant, and consequently reducing the pump suction. To make up any losses from this closed circuit, it is necessary that the coolant contained in the body of the tanks should be capable of passing into the circulation stream. Two methods of giving this condition are used:

1. By arranging the scroll outlet pipe to completely fill the header tank outlet leg, and arranging make-up holes in the lower portion of the scroll outlet pipe.
2. By arranging the scroll pipe outlet to be less than the internal diameter of the header tank outlet, and causing a small percentage of the total flow to be discharged into the body of the tank via the separation holes. This discharged water is then made up by a flow from the body of the tank, and around the scroll pipe outlet into the circuit.



■ Fig. 4 – Header tank

■ Thermostatic Header Tank Relief Valve

In the pressure cooling system the coolant jacket, header tank, radiator, and connections are hermetically sealed, so that under normal conditions the vapor generated by heating the coolant remains in the system where it exercises pressure on the liquid, raising its boiling point and therefore its capacity for absorbing heat from the engine.

In this type of system, therefore, it is necessary to provide an automatic valve which will (a) act as a safety valve to avoid excessive pressure within the cooling system, (b) admit air into the system when the internal pressure falls to approximately 2 psi below atmospheric pressure, and (c) provide a purge at all temperatures for any incondensable gas which may exist in the system causing a false pressure and a consequent loss of coolant.

At temperatures above 212 F, the valve prevents the pressure in the system from exceeding the saturated steam pressure for that temperature by more than 3½ psi, by releasing any air or incondensable gas which may be present in the system.

The valve prevents the pressure in the system from exceeding a predetermined value, normally 30 psi. In this respect it operates exactly as an ordinary spring-loaded safety valve.

On falling temperatures, the valve prevents the internal pressure from falling to more than 2 psi below the atmospheric pressure by admitting air.

A sectional view of a valve which has been used successfully is reproduced in Fig. 5.

Body – The body comprises an upper casting *O* having an outlet to the atmosphere and a lower casting *P* arranged to screw into a hole in the header tank, and having a hole *X* tapped ¼ in. BSP for connecting a pressure gage.

Valve – Although the valve performs three separate and distinct functions, only one valve and seat are employed. The valve proper is the annular ring on the plate *C* which mates with the neoprene ring *T* bonded to the plate soldered to the upper end of the bellows *E*.

Thermal Element – The thermal element consists of a seamless metallic bellows *A* soldered to the top plate *H*, valve plate *C* and a hollow metallic phial *B*. A through connection between the interior of the phial and the interior of the bellows is provided.

Bellows and phial are charged with the same fluid as is used for cooling the engine. On rise of temperature, the thermal bellows expands against the pressure of the main blow-off spring *J* which is preloaded and held in position on the spindle *U* by the retaining ring and cotters *S*.

The bottom plate *C* carries the bellows cage *D*, the top of which acts as a limit stop for the expansion of the bellows *A*. A light spring *L* is provided to exert the load on the cage *D* necessary to maintain an internal pressure of 2 psi during the warming up period.

Cage *D* around the thermal element is pierced with six 11/16-in. diameter holes.

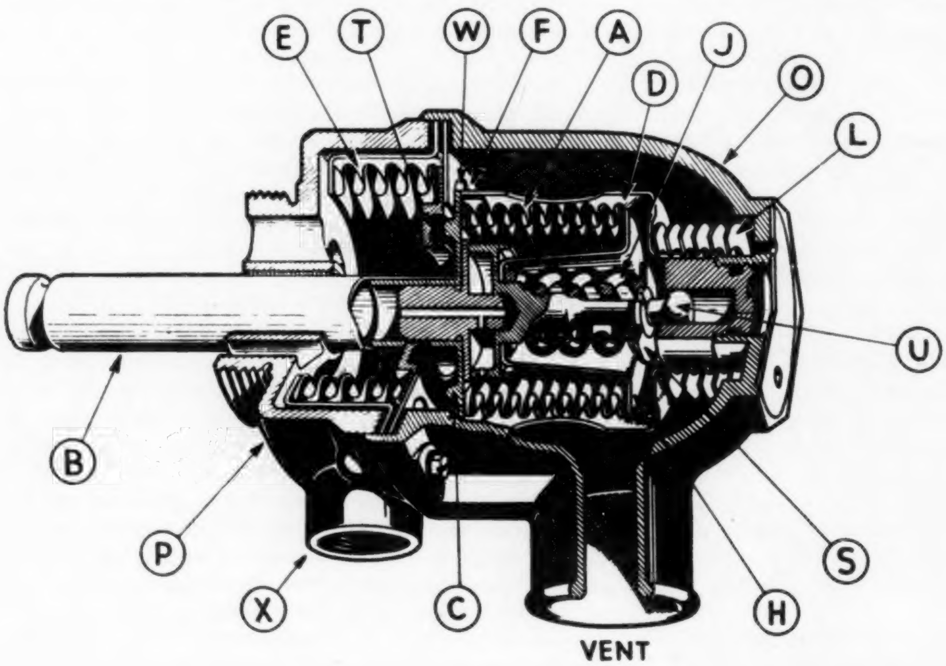
Vacuum Element – This consists of the vacuum bellows *E* which is open at its lower end to the pressure within the header tank, and attached to the plate containing the neoprene seating ring *T*. Upward movement of the vacuum bellows is limited by the stop-plate *F* which, with the vacuum bellows housing, is clamped between the top and bottom castings *O* and *P*. The stop-plate *F* also limits the downward movement of the thermal element.

Function 1 – To prevent loss by "splash" while the engine is being run up to temperature, the valve *C* is held down on its seat *T* by means of a light spring *L*. If for some reason, for example liquid expansion, the internal pressure exceeds 2 psi during this period, the valve lifts against the spring *L* to release the excess pressure. It will be noted that the valve *C*, thermal phial *B*, and the cage *D* comprise one complete unit which lifts when necessary, the thermal phial sliding in a guide.

Function 2 – The valve provides a purge for any incondensable gas or air which may be in the system, as such gas, besides occupying valuable space, may result in the internal pressure being built up to a figure higher than that represented by the saturated steam pressure for the working temperature. It will be seen that the thermal phial *B* which is charged with the same liquid as used for the engine coolant generates inside the bellows *A* the same saturated steam pressure as exists in the coolant system, and as the area of the seat *C* is equal to the effective area of the bellows *A*, the pressures are balanced and the valve *C* kept on its seat by the light spring *L*. If, however, air or some incondensable gas is present, causing the pressure on the underside of the valve seat to be greater than the pressure developed in the bellows *A*, the valve will lift and release the additional pressure when it is more than 3½ psi in excess of the saturated steam pressure.

Function 3 – The valve operates as a simple safety valve

turn to page 77



■ Fig. 5 – Thermostatic relief valve for header tank

CORROSION of BEARING ALLOYS

by L. M. TICHVINSKY

Senior Mechanical Engineer,

U. S. Naval Engineering Experiment Station,

Annapolis, Md.

FAILURES of bearings in engines caused by corrosion are of great concern to field personnel and development engineers. During wartime, bearings perform under most adverse conditions because of forced speeds, increased pressures, and "overtime" operations.

Prior to the present emergency, tin was one of the predominating constituents used in bearing alloys. The Allied Nations were deprived of practically all of the active world tin supply with the loss of the Dutch East Indies islands of Banka and Billiton. Therefore, many bearings which have been babbitted with tin-base alloys have to be lined now with various tin-substitute bearing materials.

This paper describes laboratory experiments during which corrosion resistance of various bearing alloys was determined when tested with straight, with additive-type, and with re-refined lubricating oils at elevated temperatures.

Some of the experimental engines operated with the crankcase oil temperature over 280 F and with bearing temperatures far in excess of 300 F.[1]

★ ★ ★

THE AUTHOR: L. M. TICHVINSKY, senior mechanical engineer, Internal Combustion Engineering Laboratory, U. S. Naval Engineering Experiment Station, Annapolis, Md., studied for seven years at the Prague Polytechnic Institute in Czechoslovakia. He received his Mechanical Engineering degree in 1929. Mr. Tichvinsky worked on lubrication, friction, and wear of metals in the Westinghouse Research Laboratories from 1930 to 1940, and during this period he published a number of scientific papers on bearing design, friction, lubrication, and wear. He has been in charge of bearing, friction, and corrosion investigations at the Naval Engineering Experiment Station since 1940.

Author's Note: The opinions or assertions contained in this paper are the author's and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

■ Method of Test

CORROSION of bearing alloys by lubricating oils was investigated in the Underwood oxidation apparatus (Fig. 1)[2]. The simple but effective principle of operation of the Underwood test consists of spraying hot lubricating oil onto 2-in. diameter bearing shells (Fig. 2), measuring periodically the bearing weight loss, and recording the changes in the chemical properties of the oil. The bearing weight loss is due solely to corrosion, because other losses such as those due to wiping, abrasion, and wear are not produced in the Underwood tester.

One and one-half liters of lubricating oil are placed in a tank equipped with a heater element and a thermostat control device (Figs. 1 and 2). A small gear pump circulates the lubricating oil and forces it at 10 psi through a header provided with four drilled holes. The holes are so located that four small oil jets are sprayed on the inside of four bearing shells mounted on a copper baffle-plate fixed parallel to the header. The duration of individual tests varied from 20 to 40 hr. The bearing shells were accurately weighed before test and after each 10-hr test interval, in order to record the weight loss. All tests were made at the standard Underwood temperature of 325 F. However, other higher and lower temperatures were also used during some tests. The temperature during each run was maintained constant by thermostatic control. The lubricating oil pressure was 10 psi during all runs. All tests described were made without the addition of a catalyst to the lubricating oil.

Lubricating oil analyses were made at 10-hr intervals. Half-pint samples were drawn from the oil sump immediately after each 10-hr period, while the oil was still well mixed, and replaced by the same amount of fresh oil (equivalent to the oil at 0 hr, as shown on the chemical analyses tables). The analyses were made for the determination of:

a. Viscosity at 130 F.
b. Reaction.
c. Neutralization number.
d. Carbon residue.
e. Ash content.
f. Naphtha insoluble.
g. Chloroform soluble.

The method of each analysis[3] and its significance for the interpretation of Underwood results are presented as follows:

a. Viscosity at 130 F, F.S.B. No. 304.3 (ASTM No. D 88-33). The viscosity increase, in per cent, indicates chiefly the rate of oxidation and decomposition of the lubricating oil.

b. Reaction, F.S.B. No. 510.1, shows the presence in the lubricating oil of inorganic or water-soluble organic acids or bases which were formed during the tests.

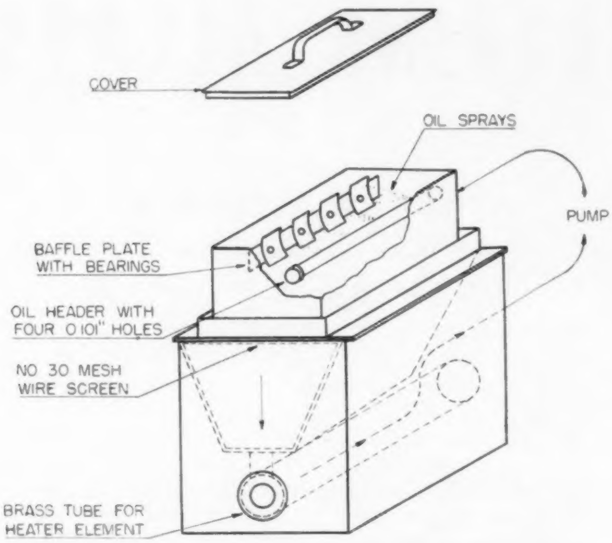
c. Neutralization number, F.S.B. No. 510.31 (ASTM

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

[1] See *Mechanical Engineering,* Vol. 64, No. 4, April, 1942, pp. 259-269: "Bearings and Lubrication," by R. J. S. Pigott.

[2] Tests to Determine the Corrosion Properties of Oils as Related to Bearing Materials," by A. F. Underwood, presented at the Summer Meeting of the ASTM, New York, 1937.

[3] See Federal Standard Stock Catalog VV-L-791a, Oct. 2, 1934, Section IV (Part 5), Lubricants and Liquid Fuels.



■ **Fig. 1 – Diagrammatic sketch of the Underwood oxidation apparatus**

No. D 188-27 T). The change in the neutralization number shows the amount of both organic and inorganic acid bodies which were formed during the oxidation test.

d. Carbon residue, F.S.B. No. 500.13 (ASTM D 189-30). The increase in the carbon residue shows the increase in the formation of asphaltic matter in the oxidized oil.

e. Ash content, F.S.B. No. 542.1. The increase in the ash content indicates an increase of corrosion which is due to the reaction of the organic acids with the bearing materials.

[4] See *Metals and Alloys*, Vol. 12, Nos. 3 to 6, Sept. to Dec., 1940: "Bearing Metals from the Point of View of Strategic Materials," by H. W. Gillett, H. W. Russell, and R. W. Dayton.

[5] See *Metals and Alloys*, Vol. 5, Aug. and Sept., 1934: "Bearing Metals of Lead Hardened with Alkali and Alkaline Earth Metals," by Leland E. Grant.

[6] See ASTM Proceedings, Vol. 41, 1941, pp. 886-893: "The Properties of Certain Lead-Bearing Alloys," by A. J. Phillips, A. A. Smith, Jr., and P. A. Beck.

[7] See *Metals and Alloys*, Vol. 15, No. 4, April, 1942, pp. 584-587: "Silver in Lead-Base Babbitts," by H. W. Gillett and R. W. Dayton.

[8] See "Bearing Metals and Alloys," by H. N. Bassett, 1937, Arnold, London.

f. Naphtha insoluble is a measure of the total inorganic matter, carbon, and asphaltic matter in the used oil.

g. Chloroform soluble is a measure of the asphaltic matter in the used oil.

■ Materials Tested

Bearing Alloys – The corrosion of the following bearing alloys was investigated:

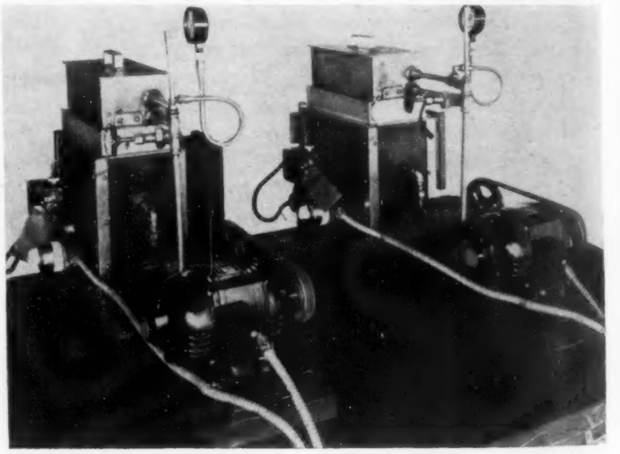
1. Tin-base babbitt.
2. Lead-base babbitt.
3. Alkali-hardened lead babbitt.
4. Arsenic lead-base babbitt.
5. Silver lead-base babbitt.
6. Cadmium-silver alloy.
7. Cadmium-nickel alloy.
8. Copper-lead alloy.

The approximate compositions of these alloys are tabulated on Fig. 3[4, 5, 6, 7, 8].

Some of the physical properties of the comparatively new arsenic lead-base and silver lead-base babbitts are as follows:

Brinell Hardness –

Babbitt	No. 4	No. 5
At room temperature	20.8	23
100 C (212 F)	14.2	..
150 C (302 F)	9.8	8.9
200 C (392 F)	5.7	..



■ **Fig. 2 (above) – Underwood oxidation apparatus**

	TIN	LEAD	ANTIMONY	COPPER	CADMIUM	SILVER	NICKEL	ARSENIC	CALCIUM	ALUMINUM	ZINC	MAGNESIUM	IRON		
NO.	SN	FB	SB	CU	CD	AG	NI	AS	CA	AL	ZN	MG	FE	OTHERS	REMARKS
1	880	035	75	40	-	-		010	-	-	-	-	0 03	BALANCE	GOOD QUALITY TIN-BASE BABBITT
2	50	800	140	050	-	-		020	-	-	-	-	-	BALANCE	GOOD QUALITY LEAD-BASE BABBITT
3	10	980	-	-	-	-	-	-	050	007	-	008	001	BALANCE	CALCIUM HARDENED LEAD BABBITT
4	10	82 50	150	05	-		-	10	-	-					ARSENIC IN LEAD-BASE BABBITT
5	20	7775	150	025	-	50		-		-				-	SILVER IN LEAD-BASE BABBITT
6	001	002	-	050	980	1 0		-	-		002	-	-	BALANCE	SIMILAR TO SAE NO 180 (CD-AG)
7	010	001	0 18	0 04	978	-	180	-	-	-	-	-	-	BALANCE	SIMILAR TO SAE NO 18 (CD-NI)
8	-	250	-	750	-	-	-	-	-	-	-	-	-	-	COARSE STRUCTURE COPPER-LEAD

■ **Fig. 3 (left) – Chemical compositions of tested bearing materials**

Compressive Strength –

Babbitt	At 5% Reduction In Length, psi No. 4	At 5% Reduction In Length, psi No. 5	At 10% Reduction In Length, psi No. 4	At 10% Reduction In Length, psi No. 5
At 25 C (77 F)	12,000	12,000	13,000	13,600
100 C (212 F)	7,600	6,800	8,800	6,800
150 C (302 F)	6,000	3,750	6,700	3,750
200 C (392 F)	3,400	....	3,800	....

Melting Range –

Babbitt	Liquidus	Solidus
No. 4	245 C (473 F)	242 C (468 F)
No. 5	259 C (497 F)	243 C (470 F)

Structure – The microstructures of the arsenic and the silver lead-base babbitts were examined, and it was found that the structures of these babbitts were essentially similar to those of commercial lead-base babbitts. The interpretations of the photomicrographs, shown on Fig. 4, are as follows: In the arsenic lead-base babbitt, the fine-grain structure (gray) is lead-antimony eutectic; the larger grain (white areas) is lead-antimony solid-solution. Interspersed in the lead-antimony eutectic are needles of copper-tin. In the silver lead-base babbitt, the background is a mixture of lead-antimony eutectic, with a few dendrites of lead-antimony solid-solution. The small needles in the eutectic are nearly pure antimony. The long needles are antimony-silver and copper-tin. The scattered cubes are antimony-tin.

Heat Conductivity – The heat conductivity of the main alloy constituent and the approximate heat conductivity of bearing alloys are expressed in per cent of the heat conductivity of pure copper[4, 9] in the following table:

1. Tin	14.8 %	Tin-base babbitt	9%
2. Lead	7.71	Lead-base babbitt	6
3. Lead	7.71	Alkali-hardened lead babbitt	5
4. Lead	7.71	Arsenic lead-base babbitt	5
5. Lead	7.71	Silver lead-base babbitt	5
6. Cadmium	23.0	Cadmium-silver alloy	21
7. Cadmium	23.0	Cadmium-nickel alloy	21
8. Copper	100	Copper-lead alloy	30

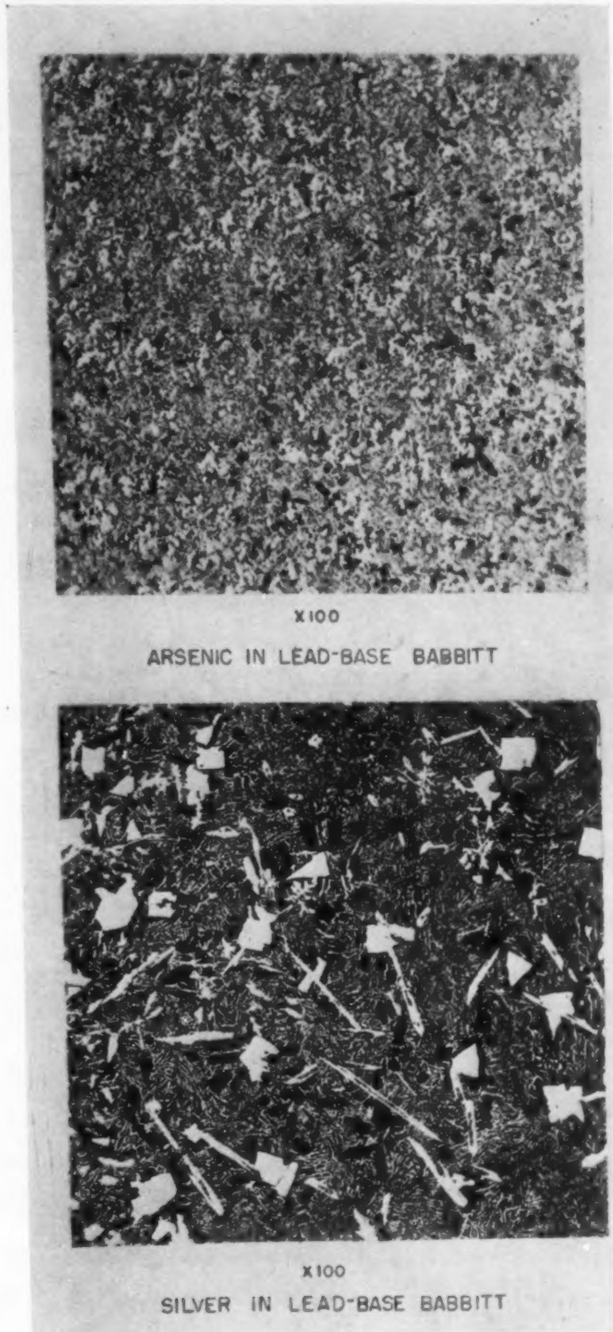
Lubricating Oils – The straight mineral and the additive-type lubricating oils used during the tests possessed various characteristics[10, 11]. For simplicity, tests with oils only of SAE 30 grades, such as Navy Symbol 2190 and Navy Symbol 3065, were used during these tests. Navy Symbol 2190 lubricating oils have a viscosity of 185 to 205 sec Saybolt at 130 F, and Navy Symbol 3065 lubricating oils have a viscosity of 60 to 70 sec Saybolt at 210 F. The chemical analyses of new and used lubricating oils are tabulated together with test results which are reported in the following section.

Test Results

A. *General* – All test results are presented on graphs on which the per cent loss in weight is plotted versus the duration of the test. Most tests were run for 40 hr at 325 F, but several 20-hr tests were also made at 350 F and 375 F.

Bearing corrosion observed was of two distinct types: uniform corrosion, which affected most of the bearing surface, and localized corrosion, which affected only isolated bearing areas, as shown in Fig. 5. The corrosion of the copper-lead bearings was mainly uniform, while the corrosion of the white bearing alloys was mostly localized.

The accelerated corrosion test results indicate that the corrosion of the tin-base bearing alloy was negligible with all lubricating oils tested. The corrosion of the lead-base bearing alloys was low and moderate, while the corrosion of the copper-lead alloys and the cadmium-base alloys was high.



■ Fig. 4 – Lead-base babbitts containing arsenic and silver (reduced from photomicrographs taken at 100 X)

[9] See Mechanical Engineer's Handbook, by L. S. Marks, p. 623: "Non-Ferrous Metals and Alloys," Third Edition, 1930, McGraw-Hill, New York.

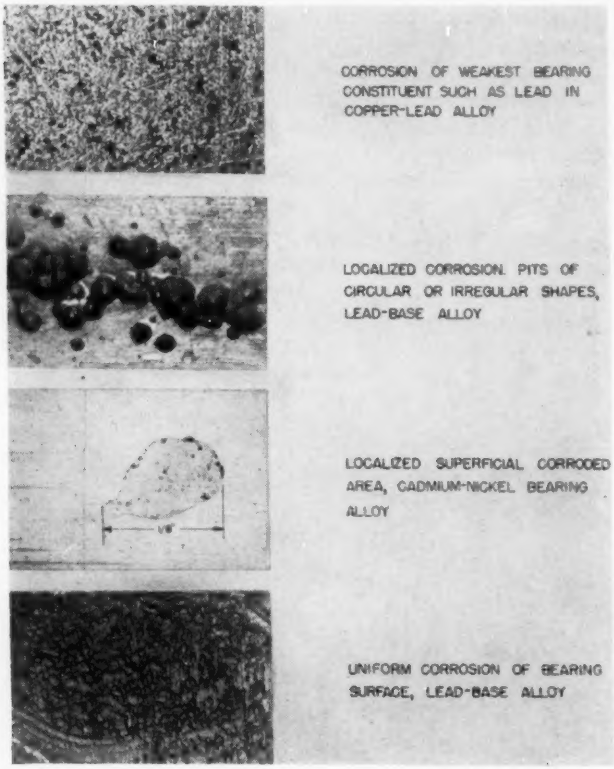
[10] See "Lubricating Oil. General Information. Requirements and Methods of Tests." Bureau of Ships, Navy Department, Feb. 1, 1942.

[11] "Aircraft Lubricating Oils for the War Effort," by C. M. Larson, presented at a meeting of the National Petroleum Association, Bradford, Pa., June 2, 1942.

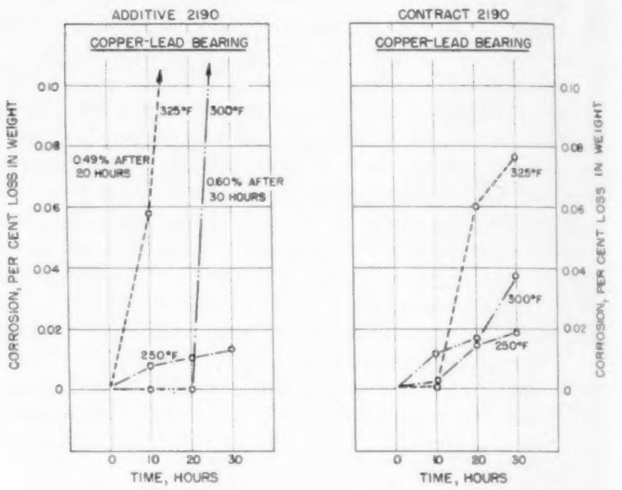
An increase in the test temperatures accelerated the oxidation and the rate of acid formation of the lubricating oil and the corrosion of the tested bearings (Figs. 6 and 7).

Corrosion tests with additive-type oils, such as those shown on these figures, will be referred to later.

The high corrosion resistance of tin-base babbitt bearing alloys is shown on Figs. 8, 10, and 12 and the corresponding oil analyses on Figs. 9, 11, and 13. Corrosion did not occur even at the highest temperatures of 350 F and 375 F (Figs. 8 and 10). This high corrosion resistance of all tin-base bearing alloys is typical and was, in the pre-war



■ Fig. 5 – Types of bearing-metal corrosion

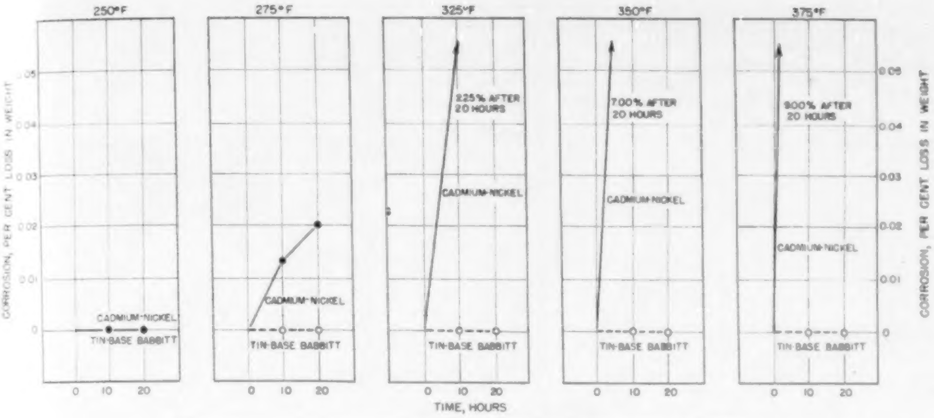


■ Fig. 6 (above) – Corrosion of copper-lead bearing material in the Underwood oxidation apparatus at 250 F, 300 F, and 325 F

LUBRICATING OIL	ADDITIVE TYPE LUBRICATING OIL								
OIL TEMPERATURE, °F	250				300			325	
HOURS	0	10	20	30	10	20	30	10	20
VISCOSITY, 130 °F	182	186	190	190	189	195	595	610	3105
VISCOSITY INCREASE, %	-	1.59	1.75	1.75	1.74	2.67	213.9	223	1543
REACTION	NEUT	NEUT	NEUT	NEUT	NEUT	ACID	ACID	ACID	ACID
NEUTRALIZATION NUMBER	0.01	0.02	0.02	0.03	0.05	0.20	20.00	16.00	31.00
CARBON RESIDUE	0.01	0.04	0.05	0.04	0.03	0.06	1.05	1.45	4.20
ASH, %	0	0	0	0	0	0	0.03	0.02	0.33
NAPHTHA INSOLUBLE	0	0	0	0	0	0	1.75	0.90	12.80
CHLOROFORM SOLUBLE	0	0	0	0	0	0	1.50	0.55	11.45

LUBRICATING OIL	CONTRACT 2190 LUBRICATING OIL								
OIL TEMPERATURE, °F	250				300			325	
HOURS	0	10	20	30	10	20	30	10	20
VISCOSITY, 130°F	200	198	200	208	234	260	295	267	500
VISCOSITY INCREASE, %		1.01		5.05	18.18	31.81	46.50	34.85	152.6
REACTION	NEUT	NEUT	ACID	ACID	ACID	ACID	ACID	ACID	ACID
NEUTRALIZATION NUMBER	0.03	0.02	0.13	0.20	1.25	2.25	2.45	1.12	6.65
CARBON RESIDUE	0.03	0.04	0.08	0.08	0.25	0.43	0.45	0.50	1.75
ASH, %	0	0	0.01	0.01	0.01	0	0	0.01	0.01
NAPHTHA INSOLUBLE	0	0	0	0.01	0.20	0.45	0.75	0.46	2.90
CHLOROFORM SOLUBLE	0	0	0	0	0	0.35	0.65	0.30	2.30

■ Fig. 7 (left) – Chemical analyses of contract 2190 and additive-type 2190 lubricating oils in the Underwood oxidation apparatus



■ Fig. 8 – Corrosion of cadmium - nickel and tin - base babbitt bearing materials in the Underwood oxidation apparatus – contract 3065 lubricating oil

CHEMICAL ANALYSES OF BEARING MATERIALS

BEARINGS	ELEMENTS							
	CU	PB	CD	SN	Sb	NI	BI	SAE NO.
CADMIUM-NICKEL	0.04	0.01	REMAINDER	0.10	0.18	1.86		18
TIN-BASE BABBITT	3.17	0.56		REMAINDER	4.72		3.99	10

period, one of the main reasons for their frequent use and application.

The low and moderate corrosions of lead-base alloys, along with the corresponding oil analyses, are shown on Figs. 10 to 15 inclusive. The least corrosive alloys in this group were the arsenic and the silver lead-base babbitts (Figs. 10, 12, and 14). The corrosion of alkali-hardened lead was low with contract 2190 lubricating oil (Fig. 14), and was excessive with contract 3065 lubricating oil, when the neutralization number rose to 13.80 after 30 hr during the test at 325 F (Figs. 10 and 11).

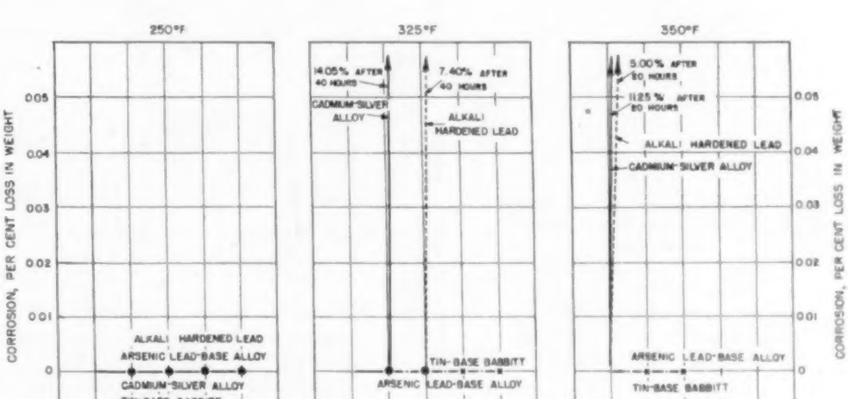
The high corrosion of cadmium-base alloys is shown on Figs. 8, 10, and 12; the chemical analyses of the oils used are tabulated on Figs. 9, 11, and 13. Excessive corrosion of cadmium-nickel occurred within the first 10 hr at 325 F with contract 3065 lubricating oil (Figs. 8 and 9). Excessive corrosion of cadmium-silver occurred within 20 hr, when the neutralization number was 4.80, and when there was no corrosion of alkali-hardened lead (Figs. 10 and 11). The excessive corrosion of copper-lead and the chemical analyses of the oils are shown on Figs. 12, 13, 16, and 17.

B. *Corrosion with Compounded and Re-refined Lubri-*

LUBRICATING OIL	CONTRACT 3065 LUBRICATING OIL										
OIL TEMPERATURE, °F	250			275		325		350		375	
HOURS	0	10	20	10	20	10	20	10	20	10	20
VISCOSITY, 130°	215	219	223	223	237	340	706	913	*	560	*
VISCOSITY INCREASE, %	-	1.89	4.23	4.23	11.3	60.6	231	328	-	163	-
REACTION	NEUT.	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID
NEUTRALIZATION NUMBER	0.01	0.04	0.15	0.25	0.83	6.5	10.0	12.25	5.10	6.75	7.95
CARBON RESIDUE	0.08	0.15	0.14	0.23	0.50	1.65	3.13	4.60	6.85	3.35	5.80
ASH, %	0	0.01	0	0.01	0.01	0.02	0.20	0.12	0.15	0.02	0.05
NAPHTHA INSOLUBLE	0	0	0	0.01	0.03	0.10	4.05	6.10	17.25	3.90	12.06
CHLOROFORM SOLUBLE	0	0	0	0	0.01	0.08	3.75	3.50	16.58	2.15	6.45

* COULD NOT BE DETERMINED AS OIL BECAME SEMI-SOLID

■ Fig. 9 – Chemical analyses of contract 3065 lubricating oil in the Underwood oxidation apparatus



■ Fig. 10 – Corrosion of various bearing materials in the Underwood oxidation apparatus – contract 3065 lubricating oil

LUBRICATING OIL	CONTRACT 3065 LUBRICATING OIL												
OIL TEMPERATURE, °F	250					325				350			
HOURS	0	10	20	30	40	10	20	30	40	10	20	30	40
VISCOSITY, 130°F	215	235	237	241	241	275	333	717	1542	953	*	-	-
VISCOSITY INCREASE, %	-	9.36	10.21	12.04	12.04	27.8	54.8	234.1	61.7	343.2		-	
REACTION	NEUT	NEUT	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	-	-
NEUTRALIZATION NUMBER	0.01	0.75	0.10	0.27	0.33	2.40	4.80	13.80	14.95	15.5	13.0	-	-
CARBON RESIDUE	0.06	0.09	0.15	0.20	0.18	0.73	1.30	2.63	3.56	3.0	4.18		-
ASH, %	0	0	0	0	0.01	0	0.01	0.03	0.13	0.09	0.11	-	-
NAPHTHA INSOLUBLE	0	0	0	0	0	0	0.37	2.64	7.50	4.8	8.9	-	
CHLOROFORM SOLUBLE	0	0	0	0	0	0	0.18	0.50	6.55	0.43	6.43	-	-

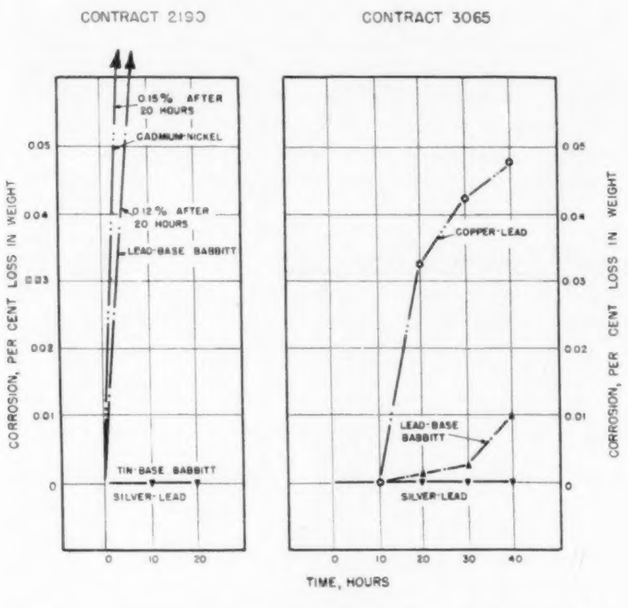
* COULD NOT BE DETERMINED AS OIL BECAME SEMI-SOLID

■ **Fig. 11 – Chemical analyses of contract 3065 lubricating oil in the Underwood oxidation apparatus at 250 F, 325 F, and 350 F**

cating Oils – Corrosion tests of various bearing materials with compounded and re-refined oils are shown on Figs. 6, 7, 16, 17, 18, and 19. The results shown on Figs. 6 and 7 indicate that the additive-type 2190 oil tested is inferior to contract 2190 straight mineral oil. Attention is called to the rapid increase of the neutralization number of the additive oil (Fig. 7). The results of tests with additive-type high-corrosive and low-corrosive lubricating oils are shown on Figs. 16 and 17. Comments relative to copper-lead corrosion are given in the succeeding paragraph. Test results, shown on Figs. 18 and 19, indicate that a satisfactory re-refined lubricating oil can be further improved and made low corrosive by mixing it with a suitable additive, Type B, as shown in Fig. 18. The same oil, however, can be made very corrosive by the addition of an unsuitable additive, Type A, as shown in Fig. 18.

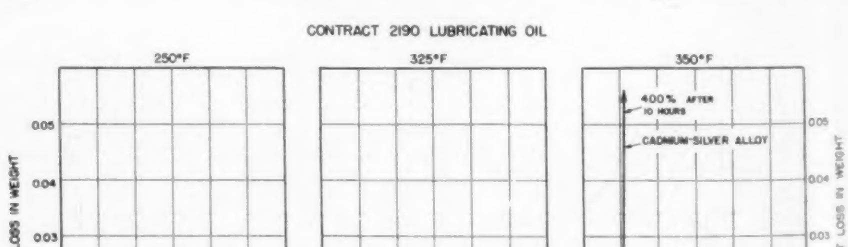
LUBRICATING OIL	CONTRACT 2190			CONTRACT 3065				
OIL TEMPERATURE, °F	325°F			325°F				
HOURS	0	10	20	0	10	20	30	40
VISCOSITY, 130°	195	490	555	215	410	750	635	365
VISCOSITY INCREASE, %	-	151	185	-	90.5	258	196	70.1
REACTION	NEUT.	ACID	ACID	NEUT	ACID	ACID	ACID	ACID
NEUTRALIZATION NUMBER	0.01	10.6	14.5	0.02	7.20	12.1	13.6	15.3
CARBON RESIDUE	0	1.75	2.05	0.05	1.86	3.00	3.25	1.90
ASH, %	0	0	0	0.01	0	0	0	0
NAPHTHA INSOLUBLE	0	3.15	4.04	0	0.50	4.25	4.75	2.75
CHLOROFORM SOLUBLE	0	3.10	4.00	0	0.20	1.85	1.83	2.70

■ **Fig. 13 – Chemical analyses of contract lubricating oils in the Underwood oxidation apparatus at 325 F**



■ **Fig. 12 – Corrosion of various bearing materials in the Underwood oxidation apparatus at 325 F**

C. *Corrosion of Copper-Lead* – Excessive copper-lead corrosion, which is shown on Fig. 16, led to a study of the corroded copper-lead specimens. Macrophotographs, shown on Fig. 20, of the new and of the two tested bearing surfaces did not indicate the cause of high corrosion. Small

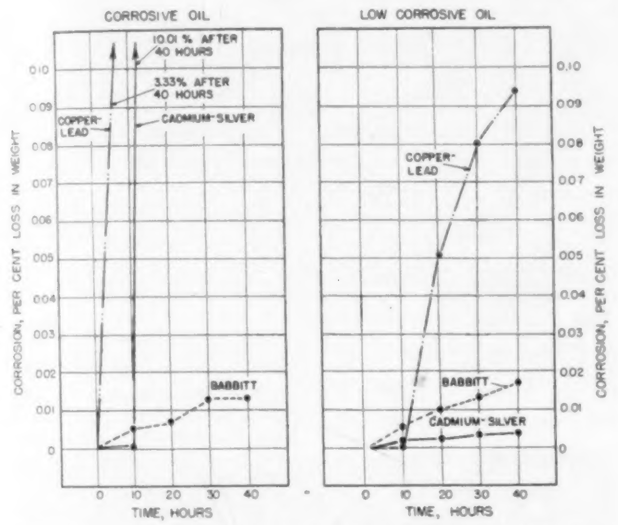


■ **Fig. 14 – Corrosion of various bearing materials in the Underwood oxidation apparatus**

LUBRICATING OIL	CONTRACT 2190 LUBRICATING OIL												
OIL TEMPERATURE, °F	250					325				350			
HOURS	0	10	20	30	40	10	20	30	40	10	20	30	40
VISCOSITY, 130°F	198	200	204	204	205	342	494	307		*		-	-
VISCOSITY INCREASE, %	-	1.25	3.56	3.85	3.60	68.6	151	105		-	-	-	-
REACTION	NEUT	NEUT	ACID	ACID	ACID	ACID	ACID	ACID		NEUT	NEUT		-
NEUTRALIZATION NUMBER	0.01	0.07	0.46	0.11	0.15	3.35	7.50	4.10	-	0.01	8.75	-	-
CARBON RESIDUE	0	0.05	0.07	0.03	0.03	0.94	1.17	0.98		0	2.95	-	
ASH, %	0	0	0	0	0	0	0	0	-	0	0.02	-	
NAPHTHA INSOLUBLE	0	0	0	0	0	1.38	2.50	1.91	-	0	4.67	-	
CHLOROFORM SOLUBLE	0	0	0	0	0	1.18	2.20	1.87		0	4.90	-	-

* COULD NOT BE DETERMINED AS OIL BECAME SEMI-SOLID

■ Fig. 15 – Chemical analyses of contract 2190 lubricating oil in the Underwood oxidation apparatus at 250 F, 325 F, and 350 F



■ Fig. 16 – Corrosion of copper-lead, cadmium-silver, and babbitt bearing materials in the Underwood oxidation apparatus at 325 F

sections of these bearings were next prepared and fixed in polishing mounts for microscopic examinations. Photomicrographs of an unused bearing section, shown on Fig. 21, indicate a typical non-solid solution structure in which particles of lead (black) were trapped during solidification between copper crystals (white). Photomicrographs of a bearing which was in operation for 40 hr with a high-corrosive oil are shown on Fig. 22. As may be seen, the excessive loss in weight was due to the loss of lead to a considerable depth from the bearing surface. The photomicrographs on Fig. 23 show the cross-section of the bearing which was in operation with low-corrosive oil for 40 hr. There was no apparent loss of lead.

The results of chemical analyses for these tests, given in Fig. 17, show that after 40 hr of test the neutralization numbers of the high-corrosive oil and of the low-corrosive oils were 9.00 and 3.00, respectively.

■ Discussion

A corrosion demerit rating, ranging from 0 to 10 maximum, which is based on the weight loss, in grams per hour, affords a satisfactory means for evaluating the overall corrosion resistance of the eight bearing materials, all of which were tested at 325 F in the Underwood oxidation apparatus (Fig. 24).

It is realized that the majority of tests were made at the rather drastic temperature of 325 F, when most of the lubricating oils begin to lose some of their essential physical properties. It was found, however, that tests conducted at the lower temperature of 250 F, for the limited duration of time allowed, offered no information for obtaining comparative data.

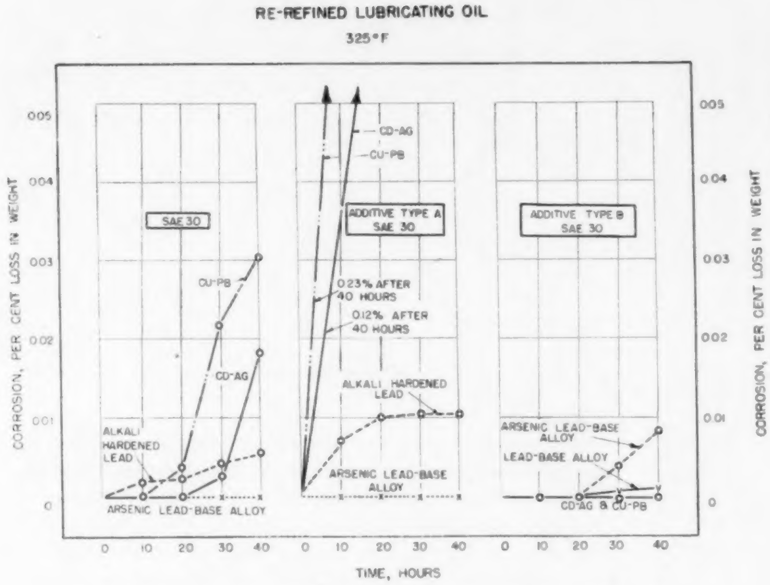
Besides direct corrosion data, these tests indicate that a careful selection of additives, if used, should be made in order to diminish the corrosiveness of the uncompounded lubricating oil[10, 11].

The Underwood oxidation apparatus proved to be a valuable tool in this investigation. Unavoidable errors during the tests were greatly reduced by repeated runs made under identical test conditions.

Interesting data were obtained in a test which was made with a new and with a used lubricating oil of the same type. The used oil, prior to the Underwood test, was in operation for 70 hr in a General Motors, model 6-71 diesel engine. The chemical analyses of the oils were recorded during a routine 40-hr test at 325 F, and are tabulated on Fig. 25. The zero column contains the chemical properties of the two oils before the test. The comparison of the

LUBRICATING OIL	CORROSIVE OIL					LOW CORROSIVE OIL				
HOURS	0	10	20	30	40	0	10	20	30	40
VISCOSITY, 130°F	200	236	450	575	425	230	243	250	325	380
VISCOSITY INCREASE, %		17.50	125	188	113	-	6.53	8.71	41.3	65.3
REACTION	NEUT	NEUT	ACID	ACID	ACID	NEUT	NEUT	NEUT	ACID	ACID
NEUTRALIZATION NUMBER	0.01	0.60	5.00	7.50	9.00	0.80	0.90	1.00	2.90	3.00
CARBON RESIDUE	0.35	1.00	2.50	3.50	4.00	0.20	0.30	0.40	1.00	1.20
ASH, %	0.10	0.10	0.30	0.30	0.30	0.30	0.35	0.35	0.35	0.35
NAPHTHA INSOLUBLE	-	0.50	3.50	5.00	5.00	0	0	0	1.20	1.40
CHLOROFORM SOLUBLE		0.20	2.80	4.50	4.80	0	0	0	0.75	1.10

■ Fig. 17 – Chemical analyses of corrosive and low-corrosive lubricating oils used in the Underwood oxidation apparatus at 325 F



■ Fig. 18 – Corrosion of various bearing materials in the Underwood oxidation apparatus

lubricated with low-corrosive oils. Adequate cooling of the lubricating oil, in this case, is of paramount importance, because it will impede the oxidation of the oil, and therefore diminish bearing corrosion.

These corrosion data should be included with such other properties as fatigue strength, compression and hardness at elevated temperatures, and heat conductivity when selecting bearing materials for various heavy-duty applications[4, 12, 13, 14, 15].

■ Acknowledgment

Acknowledgment is due to Com. E. C. Forsyth and Lt.-Com. W. C. Latrobe, Bureau of Ships, Navy Department, Washington, D. C., for permission to publish this paper. The writer is indebted to Com. A. F. Folz, superintendent, Internal-Combustion Engine Laboratory, U. S. Naval Engineering Experiment Station, Annapolis, Md., for his interest in connection with the preparation of the paper. The cooperation of W. C. Stewart, head of Metals Laboratory, U. S. Naval Engineering Experiment Station, Annapolis, Md., in obtaining physical properties of materials tested, is thankfully recognized.

LUBRICATING OIL	SAE 30					ADDITIVE TYPE A SAE 30					ADDITIVE TYPE B SAE 30				
OIL TEMPERATURE, °F	325°					325°					325°				
HOURS	0	10	20	30	40	0	10	20	30	40	0	10	20	30	40
VISCOSITY, 130°	230	275	320	370	415	230	275	310	340	420	240	250	265	275	295
VISCOSITY INCREASE, %	-	19.7	39.2	60.6	79.8	-	19.7	35.1	47.7	82.7	-	5.84	10.4	14.6	22.1
REACTION	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	ACID	NEUT.	NEUT.	NEUT.	NEUT.	NEUT.
NEUTRALIZATION NUMBER	0.25	1.00	3.05	4.75	5.60	0.25	1.75	3.15	2.75	2.50	0.11	0.23	0.21	0.18	0.70
CARBON RESIDUE	0.40	0.40	1.55	2.11	2.25	0.40	1.00	1.30	1.75	2.30	0.50	0.55	0.60	0.62	0.95
ASH, %	0	0.15	0	0.02	0.02	0	0.02	0.04	0	0	0.13	0.12	0.15	0.16	0.14
NAPHTHA INSOLUBLE	0	0.30	0.36	0.55	1.07	0	0.35	0.35	0.60	0.89	0	0.17	0.24	0.21	0.29
CHLOROFORM SOLUBLE	0	0.15	0.30	0.09	0.55	0	0.25	0.25	0.30	0.59	0.	0.15	0.22	0.08	0.18

■ Fig. 19 – Chemical analyses of re-refined lubricating oils in the Underwood oxidation apparatus at 325 F

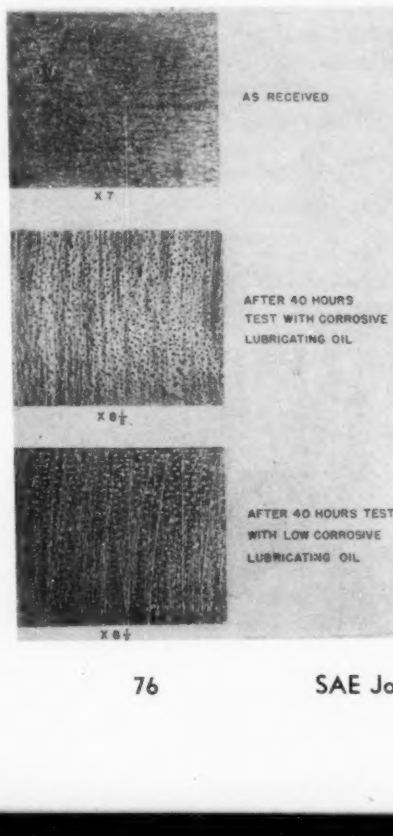
chemical properties of the new oil at the end of the 40-hr test with the chemical properties of the used oil before the test is indeed interesting. It shows that the oxidation of the new oil in the Underwood apparatus, during this 40-hr test at 325 F, was equivalent to the 70-hr oxidation of the same oil in the engine.

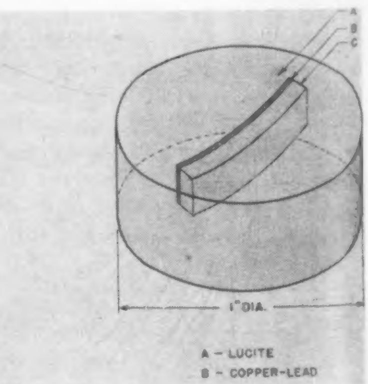
■ Conclusions

Described tests confirmed fully the high corrosion resistance of tin-base babbitt.

Lead-base babbitts containing arsenic and silver were found to possess low-corrosive properties. This indicates the practicability of continued metallurgical research for developing tin-substitute bearing materials.

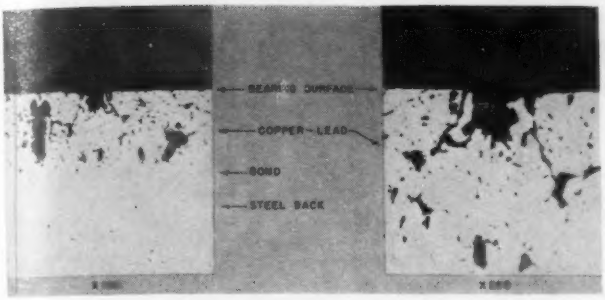
The bearing alloys which have a tendency to corrode at elevated temperatures should be



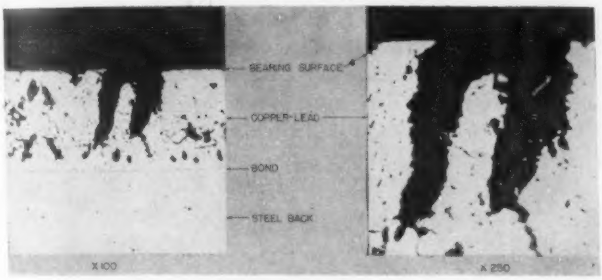


■ Fig. 20 – Macrophotographs of copper-lead bearing surfaces and sketch of bearing section in polishing mount (macrophotographs reduced)

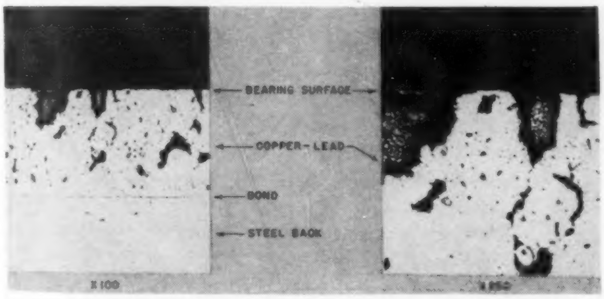
[12] See ASME Transactions, 1930, MSP-52-7, pp. 87-99: "A Study of Tin-Base Bearing Metals," by O. W. Ellis and G. B. Karelitz.
[13] See *Mechanical Engineering*, Vol. 64, No. 6, June, 1942, pp. 439-448: "Bearings for Diesel Engines," by A. B. Willi.
[14] "Diesel Engines for the Navy," by Lt.-Com. M. M. Dana, presented at the Annual Meeting of the SAE, Detroit, Mich., Jan. 9, 1941.
[15] See ASME Transactions, 1932, IS-54-3, pp. 11-24: "Straight Copper-Lead Alloys versus Leaded Solid-Solution Bronzes for Heavy-Duty Bearings," by F. R. Hensel and L. M. Tichvinsky.



■ Fig. 21 – Copper-lead bearing as received (reduced from photomicrographs taken at 100 and 250 X)



■ Fig. 22 – Copper-lead bearing after 40 hr of test with high-corrosive lubricating oil — 325 F (reduced from photomicrographs taken at 100 and 250 X)



■ Fig. 23 – Copper-lead bearing after 40 hr of test with low-corrosive lubricating oil — 325 F (reduced from photomicrographs taken at 100 and 250 X)

Testing and preparation of data were done with the assistance of P. G. Kestler, junior mechanical engineer, and C. H. Grant, junior engineering aide, both of the U. S. Naval Engineering Experiment Station, Annapolis, Md.

NUMBER	BEARING MATERIALS	RATINGS GRAMS PER 1 HOUR	NUMBER OF TESTS	ORDER OF MERIT
1	TIN-BASE BABBITT	0.001	17	I
2	LEAD-BASE BABBITT	0.004	7	III
3	ALKALI-HARDENED LEAD	0.427	50	IV
4	ARSENIC LEAD-BASE BABBITT	0.002	13	II
5	SILVER LEAD-BASE BABBITT	0.002	4	II
6	CADMIUM-SILVER ALLOY	1.724	62	VI
7	CADMIUM-NICKEL ALLOY	2.424	4	VII
8	COPPER-LEAD ALLOY	0.453	65	V

■ Fig. 24 (left) – Corrosion demerit ratings

A Brief Survey of the Principles of Pressure Water Cooling

continued from page 68

to avoid excessive pressures in the system, the valve normally being set to blow off at 30 psi. At temperatures in excess of 212 F, the bellows *A* is expanded until the top plate *H* strikes the cage *D*, which acts as a stop; thus the valve is no longer balanced and acts as a conventional spring-loaded safety valve blowing off against the pressure of the main spring *J*.

Function 4 – If the cooling system is suddenly cooled, the internal pressure falls and a partial vacuum is formed. At a pressure of 2 psi below atmospheric, the vacuum bellows *E* collapses and withdraws seat *T* from contact with the valve *C*, which is prevented from descending by the stop *F*.

Air is now admitted into the system through the holes *W* until the internal pressure is increased to a minimum of 2 psi below atmospheric pressure. During normal working, the vacuum bellows *E* are extended by the pressure and thus the valve seat *T* is definitely located and sealed against the stop plate *F*.

■ Conclusion

This short review is not intended to cover the subject of pressure cooling for liquid-cooled aircraft engines in anything more than a general manner, but as it is based on the author's experience over a lengthy period of liquid-cooled engine development, it is hoped that it may serve the purpose of arousing interest in the subject. As the speeds and operational altitude of modern aircraft continue to increase, this subject will become of more and more importance to installation engineers.

CHEMICAL DATA		HOURS 0	10	20	30	40
VISCOSITY, 130°	NEW	246	261	291	287	302
	USED	296	876	847	797	890
VISCOSITY INCREASE, %	NEW	0	69.5	19.3	17.6	23.0
	USED	21.3	20.4	19.4	17.7	20.9
REACTION	NEW	NEUT	NEUT	NEUT	NEUT	NEUT
	USED	NEUT	ACID	-	ACID	ACID
NEUTRALIZATION NUMBER	NEW	0.20	0.55	0.88	1.23	1.40
	USED	1.35	9.40	-	8.9	14.6
CARBON RESIDUE	NEW	0.15	0.21	0.32	0.53	0.74
	USED	0.98	4.50	4.84	4.09	3.87
ASH, %	NEW	0.12	0.13	0.13	0.16	0.17
	USED	0.18	0.27	0.31	0.38	1.80
NAPHTHA INSOLUBLE	NEW	0	0	0	0	0
	USED	0.86	4.70	10.9	8.40	9.20
CHLOROFORM SOLUBLE	NEW	0	0	0	0	0
	USED	0.16	2.80	7.80	-	3.70

USED OIL - WAS IN OPERATION FOR 70 HOURS IN A 6-71 DIESEL ENGINE

■ Fig. 25 (above) – Chemical analyses of new and used lubricating oils tested under identical conditions in two Underwood oxidation apparatus at 325 F

SUBSTITUTE MATERIALS—Have

THE problem of providing substitute materials for products which have been in use for a long period of time and giving satisfactory service, but require materials that have become scarce or even forbidden, has become of primary importance to all manufacturing industries. The problem must be met by the engineer, by the purchasing agent, and by the production man, as it is not a question of design only, but of availability and manufacturing complications.

Substitutions can be made by simply changing the material specification on the drawing and releasing it, which is the most dangerous method. The service difficulties which can arise from such a method might later present a real problem – much larger than the original one of substitution.

The thorough method, of course, is by the process of elimination through long periods of testing, redesign, and exhaustive research, which when completed, may produce a substitute that is just as satisfactory as the original or may even give longer life than the original design. This method, however, cannot be employed in all cases on account of the amount of time required. In the present emergency, production cannot wait!

Therefore, the engineer must analyze the problem, dividing the parts in different groups, such as cooling system, powerplant, safety items, and comfort items; and then establish a policy for attacking each group. In establishing this policy, the study should consider the consequence of failure, whether it would cause the vehicle to cease operation or whether it would only cause an inconvenience; whether it would cause a wreck, in the case of a safety item failing, or whether it would merely get the driver's feet wet, in the case of a weatherstrip failure. Some items still require extensive tests to prove whether substitutes can be released or not, and other items can be decided upon in the drafting room; in other words, both methods of providing substitutes, previously described, can and should be used.

Some specification standards which have been arbitrarily set up are on the safe side; this has been done to satisfy occasional extreme climatic conditions presented by nature that must be met by the engineers. In the case of cooling, radiators with steel fins do not provide as efficient cooling as radiators with copper fins even though the fan capacity is increased and a fan shroud employed. However, one means of making this type of core perform satisfactorily is to increase the boiling point of the coolant by using a pressure cap. The result is a cooling system that operates at a higher temperature but still meets requirements in the desert satisfactorily.

Throughout the industry, it has been the policy not to sacrifice durability of items involving safety. When substitution of material became necessary for forgings in steering knuckles, steering arms, and other axle and steering parts, exhaustive tests were made to obtain equally good or better life with the substitute materials, so that no failure of parts was invited which would involve the lives of men. The same was true of brake parts. Changes in radii or fillets on some parts, applied in such a manner as not to affect interchangeability, often made the substitute steel satisfactory, where substitution alone without modification of design did not produce an acceptable result. This could be attributed to the somewhat different physical characteristics of the substitute steel.

New regulations to restrict road speed have favored an increase in life of parts in passenger cars on the average; such is not true, however, in trucks to any extent, as the restricted speeds are not much lower than the average speed prior to the new regulations.

Numerous suggested substitute materials have not always worked out satisfactorily in regard to availability as planned. This is due to the sharply increased demand for the new material when nearly everyone changes over to it at the same time, or due to the miscalculations of available supply. Substitute materials, which appear to be available in sufficient quantities at one time, may become so scarce when released for production that immediately we are forced to find another substitute. After this has been repeated several times, it soon becomes apparent that one or more of these earlier materials which were dropped again becomes available due to the highly fluctuating demand. It is then a question as to whether that material should be tried again or the search continued for a material having more certain stability of supply.

Crude rubber, for example, was first replaced by using reclaimed rubber in a majority of cases; then, in the endeavor to reduce the consumption of this latter material, other materials such as cotton webbing and jute were used. In a short time these materials became critical, and other materials such as felt, wood, paper, plastics, and steel were resorted to. One of the natural substitutes for nickel was chromium, but chromium became critical with the isolation of Turkey. Other alloying elements then came into the picture such as manganese, vanadium, tungsten, and molybdenum. The outcome of this was the formulating of the National Emergency steels. These substitutions are brought about by successful conquests of the enemy, cutting off our sources of supply; tin and rubber from the East Indies, burlap and manila hemp from the Philippines, chromium from Turkey, tungsten from China and Burma, mica and jute from India, and so forth. Other substitutions for domestic material have been brought on by the exceedingly heavy demands of the war production program; this applies to copper, aluminum, zinc, cotton, wool, wood, and steel.

At this point, it is interesting to note why these materials have become critical. They can be classified into several

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 13, 1943.]

[1] Assistant chief engineer, Chevrolet Motor Division, General Motors Corp.

[2] Engineer in charge of material analysis, Chevrolet Motor Division, General Motors Corp.

We Gone the Limit?

by

JOHN G. WOOD[1] and R. F. SANDERS[2]

THIS paper discusses various methods employed in making substitutions, and how each substitution is a study in itself and therefore must be handled individually. A typical example, such as a truck steering knuckle, is followed through the series of tests conducted to obtain a satisfactory substitute steel. It is also pointed out that in several cases satisfactory substitute materials became unavailable, and the development procedure had to be continued towards an available material.

The development of substitute materials has brought to light other changes which aid the use of these lower grade materials, such as slight changes in design, increased radii, addition of fillets, improved machining, use of shot blasting, and improved carburizing materials.

Have we reached the limit of substitutions? The answer is definitely no, regarding the production of war materials. The extent to which we may be forced to go will depend upon availability of materials and the length of the war. Regarding postwar conditions, the answer will be provided by the availability of materials and economic reasons based on relative costs.

* * *

THE AUTHORS: JOHN G. WOOD (M '11) former vice-president representing the Passenger Car Engineering Activity of the SAE, is assistant chief engineer of the Chevrolet Motor Division of General Motors Corp. He became affiliated with General Motors about 16 years ago when he left his position as president and general manager of the Allied Magnet Wire Corp. to become executive engineer of Olds Motor Works. For a short period he was general manager of the Muncie Products Division, returning to Olds as chief engineer. He received his B.S. degree from Purdue University in 1907, and his M.E. degree in 1910, also from Purdue. Mr. Wood joined Chevrolet in 1933. R. F. SANDERS (M '40), Chevrolet engineer in charge of material analysis, design control and statistics, received his engineering degree from Michigan State College in 1929. For the next five years he was in the Engineering Department of the Olds Motor Works at Lansing as a junior engineer, first in records and then on special assignments, mainly on chassis and cooling developments. In 1934 he became a special test engineer for Chevrolet at the General Motors Proving Ground, and in 1940 was transferred to the Chevrolet Central Office in Detroit.

groups, as follows: One group contains rubber, tin, chromium, tungsten, hemp, cork, and India mica. This group has become critical due to the principal sources of supply being in the hands of the enemy or the shipping lanes between those sources and us being controlled by the enemy.

Another group, containing copper, brass, bronze, copper alloys, zinc, manganese, nickel, lead, and aluminum, has become critical due to the shortage of labor in mining compared to the exceedingly high demand for the material. The principal sources are still in our possession, and it is our problem to produce the materials in large enough quantities to meet the demands of the war production program, and to conserve as much as possible where the use is not considered essential.

A third group became critical in spite of the source being in our possession and the supply of labor being ample. This group is made up of the synthetic rubber and thermoplastic materials. The restriction in this group is manufacturing facilities, which were not inaugurated soon enough. Of course, the loss of our principal source of rubber supply to the enemy put the heavy demand on this group of materials.

The steel group, however, is critical due to various reasons. The principal source of supply is still in our possession and the labor supply, although not plentiful, cannot be considered a reason for steel being critical. The very nature of this war, being mechanical, brings an unprecedented demand for steel and more steel. The complexity of shapes and sizes, and many alloys and grades that are required, make the problem highly involved. To increase the output of steel products not only requires more steel for the added production material, but more steel for added furnaces, machinery, equipment, tooling, and transportation. Substitutions for materials in the previously mentioned non-ferrous groups also usually result in the consumption of more steel.

In making substitutions for a critical alloy steel used in a "safety item," such as a steering knuckle, we must first establish a base line for future comparisons. Up to the time of mass substitutions, our principal source of testing for such items was road testing. A standard road test over a fixed durability schedule on our proving ground consisted of 20,000 miles, with frequent periodic inspections. When the test was repeated on several vehicles and final inspection revealed no questionable characteristics, the item was released for production. Although testing was stopped at this point, observation was not. Through our field organization and systematic reports, we were informed immediately of any failures. Perhaps there were unusual conditions under peculiar circumstances that produced these failures, but any failure of a safety item is serious and warrants immediate correction.

It is obvious that such a complicated and time-consuming procedure could not be used repeatedly for the judging of substitute materials. However, we could take advantage of our past field experience with these safety items. They had been in service for a number of years and their record of performance was established. It was then a simple step to set up in the laboratory an accelerated test which would

cause a failure of the part in a comparatively short time using normal loads, but applying them and releasing them much more frequently than would be normal in service. This is the basis for the "stroking test."

After establishing definite cycles for failure on this stroking test, it is apparent that material comparisons can be made quickly, and if the results of the test on a particular substitute are as good or better than the production material, then it is safe to use it in production. Inversely, if the results are inferior to the established base line, then that material must be rejected.

Fig. 1 shows a battery of stroking machines as installed in our test laboratory.

The load applied is measured by measuring the deflection of a ring built in the piston rod; however, instead of reading the number of thousandths of an inch deflection, the gage is calibrated directly in lb load. This set-up is shown in Fig. 2.

The stroking machine can be used in a large number of applications; for example, in Fig. 3, we apply the stroking load at the end of a rigid I-beam lever which multiplies the load on a rear-axle assembly.

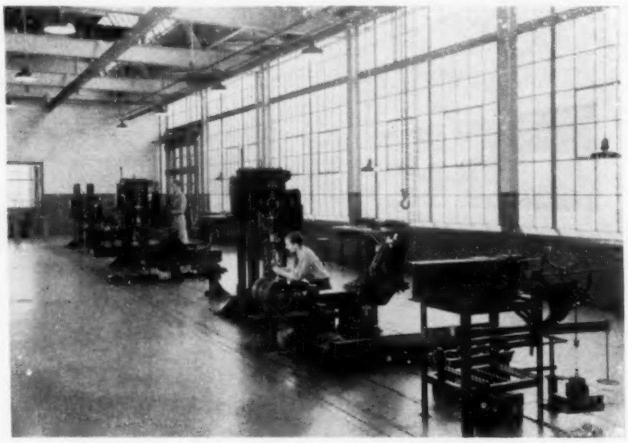
Fig. 4 shows a close-up view of a steering and third arm mounted on a support plate with the load applied to the steering ball.

Figs. 5 and 6 are views of the inspection table displaying parts which have failed in the stroking test. Test data accompany each specimen.

A good example of our laboratory procedure and the amount of engineering work involved can be brought out by the long series of tests conducted to obtain a satisfactory substitute steel for the $1\frac{1}{2}$-ton truck steering knuckle, Chevrolet Part No. 596571.

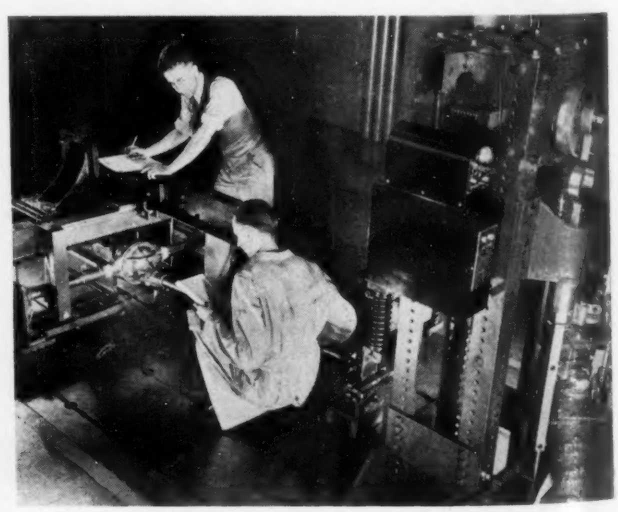
The test set-up is shown in Fig. 7. The steering knuckle with the spindle pointed vertically upward was mounted to an anchor block fastened securely to the floor bed plate. A pin (representing the king pin) was driven through the king-pin hole of the steering knuckle and aligning holes which had been drilled in the anchor block, and cap screws were placed in the knuckle apron mounting holes to prevent it from tipping on the pin. A $1\frac{1}{2}$-ton truck front wheel hub assembly with bearings, was installed on the spindle. A loading arm of 15.92-in. effective length (equal to rolling radius of $1\frac{1}{2}$-ton stake truck) was bolted at one end to the hub at the wheel bolt holes, and the other end was connected to a spring saddle attached to a stroking machine piston.

Various trial stroking loads were used in preliminary tests on production steering knuckles obtained from stock. For these preliminary tests the load was successively decreased until failure was produced with a minimum of 100,000 cycles at 260 cpm, each cycle being the application of the load first downward and then upward from the horizontal position of the loading arm. From these checks, the load of 783 lb (representing a front wheel weight of 1305 lb $\times$ 0.6 coefficient of friction) was selected. Several samples of each type steel were stroked until failure to obtain low, high, and average values for fatigue lives. The following program was carried out on the above mentioned steering knuckle.



■ Fig. 1 – A battery of stroking machines

■ Fig. 2 – Set-up for measuring the load



■ Fig. 3 – Testing a rear-axle assembly

Laboratory Fatigue Tests on Stroking Machine

Test No. 1

Material	No. Cycles to Failure	
X3140 (Ni-Cr) Original Specifications	128,000	Low
	241,000	High
Consistency (High:Low)	159,000	
Ratio 1.88:1		
	176,000	Average

Test No. 1 established the laboratory base line on production material which had proved satisfactory in field service. Therefore, 128,000 cycles for failure was taken as the absolute minimum acceptable. The ratio of the low and high fatigue lives was used as a means of measuring the consistency of the material. The ideal ratio would, of course, be 1:1.

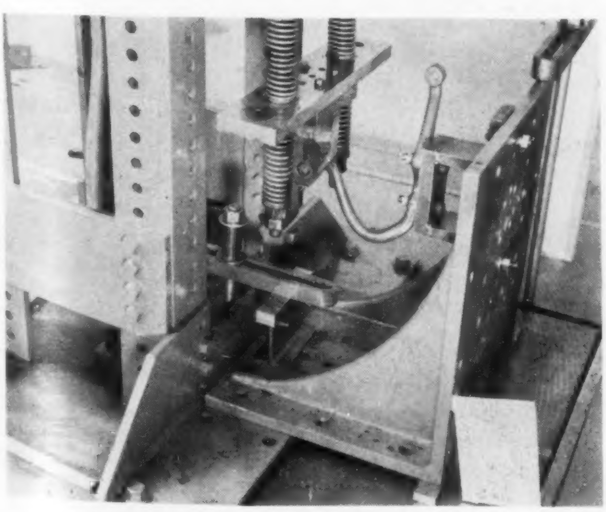
Test No. 2

Material	No. Cycles to Failure	
C-4140 (Moly) First Substitute	188,870	Low
Consistency (High:Low)	365,000	High
Ratio 1.93:1	230,000	
	261,290	Average

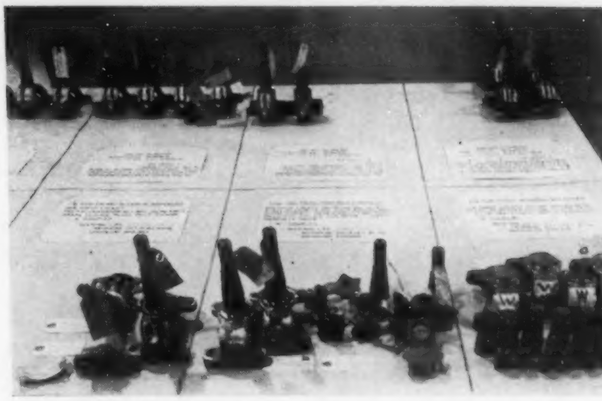
Test No. 2 showed C-4140 to be a satisfactory substitute for X3140 and was therefore released for production. The supply of C-4140 was not adequate to meet our demands, and accordingly further work was necessary to develop another substitute.

Test No. 3

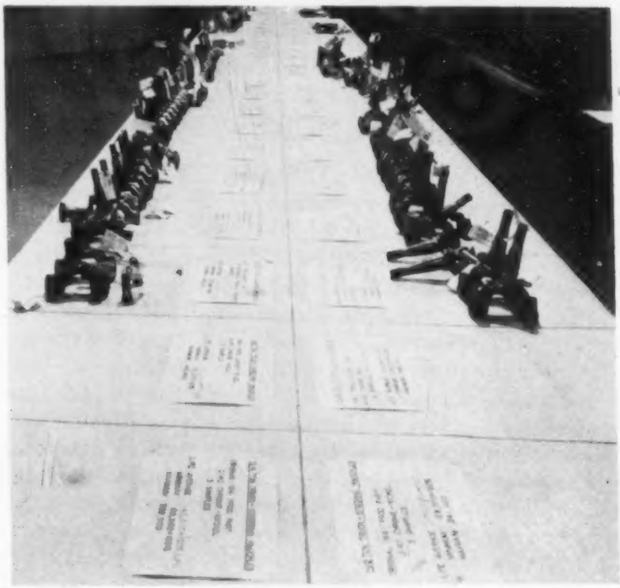
Material	No. Cycles to Failure	
Manganese – Grainal Treated	114,920	Low
Consistency (High:Low)	189,600	High
Ratio 1.65:1	187,000	
	128,637	
	155,039	Average



■ Fig. 4 – Mounting of steering and third arm – load applied to steering ball

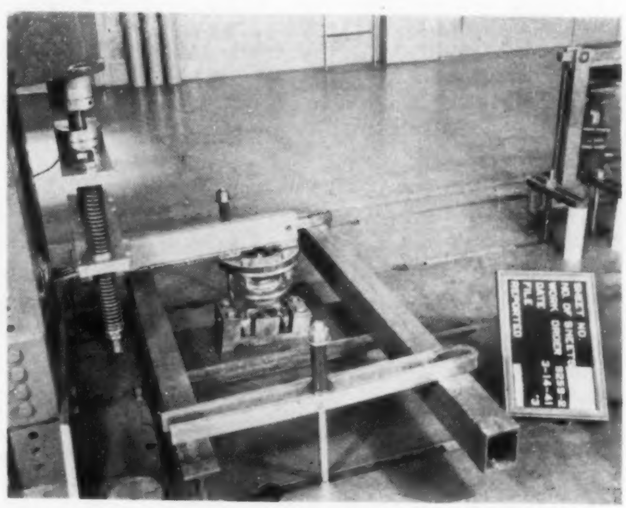


■ Fig. 5 – Inspection table showing parts that have failed



■ Fig. 6 – Inspection table showing parts that have failed

Test No. 3 showed manganese-Grainal-treated steel to be unsatisfactory when comparing the low, high, or average data with Test No. 1.



■ Fig. 7 – Set-up for testing a steering knuckle

Test No. 4

Material	No. Cycles to Failure	
MG♯10 – Grainal Treated	143,942	Low
Consistency (High:Low)	276,600	High
Ratio 1.93:1	157,462	
	194,500	
	193,126	Average

Test No. 4 showed MG♯10 – Grainal-treated steel to be satisfactory, but the apparent supply of this material vanished as soon as our laboratory tests were completed, causing the search for a satisfactory substitute to be continued. Other tests of this material were not so encouraging due to poor hardening characteristics, and it was not felt advisable to use it even though this test showed good results.

Test No. 5

Material	No. Cycles to Failure	
MG♯1 – Grainal Treated	91,411	Low
Consistency (High:Low)	526,650	High
Ratio 5.77:1	371,190	
	517,287	
	376,635	Average

Test No. 5 showed such a wide spread between high and low fatigue life values that the average showed a distorted picture. Note the extreme high-low ratio. This material was not considered satisfactory in view of the one low value.

Considerable experience in observing the location and nature of the cracks had been obtained up to this time and accordingly it was decided to increase the size of several fillets and various radii to relieve localized points of stress. Therefore this same material was tested having a larger radius at the flange plate pilot (Test No. 6).

Test No. 6

Material	No. Cycles to Failure	
MG♯1 – Grainal Treated	190,076	Low
Consistency (High:Low)	606,897	High
Ratio 3.19:1	365,991	
	339,000	
	375,491	Average

Test No. 6 showed a considerable improvement by a very slight change in design – compare the low and high data with Test No. 5, the average figure in Test No. 5 is distorted, as explained above, due to the extreme variations in test results.

It should also be noted that the results in Test No. 6 are approximately 213% of those obtained on the original material in Test No. 1.

Test No. 7

Material	No. Cycles to Failure	
MG♯1 – Grainal Treated	231,679	Low
Consistency (High:Low)	432,050	High
Ratio 1.87:1	416,650	
	305,460	
	346,460	Average

Same as Test No. 6, but an increased radius as well as a collar were added at the flange plate pilot; also quality of machining was improved at these points.

Although the average figure was no better than was obtained in the two previous tests, the spread between the high and low was considerably less, and what was considered more important was the fact that the low figure was by far the best one obtained up to this time. Note that the high-low ratio is approximately the same as obtained in Test No. 1 on production material.

Also, it is interesting to note that in Test No. 5 we stated that this material was not considered satisfactory in view of the one low value. Now in Test No. 7 we state that the results are the best obtained up to this time. Both tests were run on the same material, this difference being only a slight change in design by increasing the radii of the fillets and improving the quality of the machining in the fillets; these changes stayed in the picture after Test No. 7.

At this point in our program, we made many tests from sample production runs, comparing hard and soft parts; we also experimented with water and oil quenching. Tests showed practically no difference between knuckles that were quenched in oil and those quenched in water.

The supply of MG♯1 steel was not to be depended upon, also we wished to get a satisfactory National Emergency steel available, so the program was continued in that direction.

Test No. 8

Material	No. Cycles to Failure	
NE 8447 (No Nickel or Chromium)	147,659	Low
	695,000	High
Consistency (High:Low)	389,123	
Ratio 4.71:1	376,550	
	267,593	
	217,000	
	348,821	Average

Although the NE 8447 steel showed a wide spread between low and high values, this spread as well as the average value was higher than the original X3140 (nickel-chrome) material in Test No. 1, and was satisfactory. No metallurgical irregularity could be found to account for the wide spread in fatigue lives of this material.

To further the study of NE steels, we obtained samples of NE 8744 (containing small amounts of nickel and chromium, namely, 0.5%).

Test No. 9

Material	No. Cycles to Failure	
NE 8744	124,400	Low
Consistency (High:Low)	416,561	High
Ratio 3.34:1	307,000	
	224,517	
	198,738	
	169,292	
	373,000	
	142,906	
	244,552	Average

Although the average and high values of the NE 8744 steel were better than those obtained with the original production X3140 material, the low value was under that shown in Test No. 1 (128,000).

This material may have been satisfactory in production, but we felt that it was borderline; and in the attempt to

improve it we obtained samples of NE 8744 that were shot blasted at all areas subject to stress concentrations.

Test No. 10

Material	No. Cycles to Failure	
NE 8744 (Shot Blasted)	639,000	Low
Consistency (High:Low)	1,636,980	High
Ratio 2.56:1	1,162,425	
	1,146,135	Average

The fatigue lives of the shot blasted material were far greater than for any of the materials tested and showed that great benefit is derived from this operation.

Summing up the foregoing tests, it was possible to develop steering knuckles of much less critical material than originally used and having a greater fatigue life than original production.

The minor design changes and improvement in machining the fillets were a major help in improving fatigue life.

Although shot blasting improved fatigue life greatly, no production equipment was available for this application, and since combinations without shot blasting were found satisfactory this operation was not used in production for the manufacture of steering knuckles. It was used, however, in the manufacture of transmission gears.

Transmission gears were originally made of SAE 4815 steel, which has no chromium but ranges in nickel content from 3.25 to 3.75%. By testing production material on the dynamometer, 40 hr minimum on a certain test was established as the base line. These tests showed CX-4120 steel (no nickel and only 0.60 to 0.80% chromium) to be satisfactory. However, when put into production, CX-4120 proved to be erratic inasmuch as failures ranged from 6 hr to over 40 hr on dynamometer tests. Numerous changes in processing and heat treatment proved to be of little value. Shot blasting was tried and immediately the fatigue life was improved to well over the 40-hr minimum.

Equipment was installed and now these gears are shot blasted in production. It was found that NE 8620 steel could be used as a satisfactory substitute if it also was shot blasted.

Another aid in the development of substitute materials was the use of a non-burning type of carburizer containing no charcoal. The original steel used in a certain ring gear was of 4320A analysis; substitutes made of CX 4120 and NE 8724 were developed satisfactorily after improving the tooth contacts; however, unless shot blasting was resorted to NE 8124 could not be made to pass the dynamometer tests until the new type non-burning carburizer was used. The results obtained with this material were satisfactory, deeper penetration was produced and with greater uniformity.

To produce this material, coal is pulverized and the energizing chemicals mixed thoroughly through it. The mixture is then sintered into briquettes at 2000 F in a carefully controlled atmosphere. The briquettes are then broken into kernels of the proper size suitable for various carburizing operations. This produces material that does not fire as easily as charcoal and is longer lived due to retaining its potency even after being broken up. The energizing chemicals are an intimate part of the kernels, which is not true of previously used charcoal carburizers having the energizing chemicals coated on the outside of the kernels.

It becomes apparent in dealing with various substitute material programs that certain limitations are finally reached which necessitate the acceptance of shorter life in some units or parts. For example: in the case of the composite steel and copper radiator described earlier, it was possible to obtain a satisfactory result from a cooling standpoint, but the durability of the unit could never be made equal to the original design. The tubes carrying the water are of copper, therefore equally as good as the original; however, the cooling fins, being made of steel, must be processed with some form of rust preventative. The location of a radiator on the average truck places it in a position to be showered with sand or flying gravel, which will soon remove much of the protective coating with a consequent failure due to rusting away of the thin gage cooling fins. It should be borne in mind, however, that a failure of this type is a gradual one and does not occur suddenly; therefore ample warning is given for replacement.

Another form of limitation is reached in items such as drain cocks, where no satisfactory substitution of material that will prevent freezing of the working parts has been found within reasonable commercial limits; therefore, it resolves itself to the substitution of pipe plugs in place of drain cocks or the continuation of the original material specifications.

In the electrical units there seems to be no alternative to the use of copper. Any material which has the equivalent electrical characteristics of copper reaches into the field of precious metals such as silver and platinum, which are obviously scarcer than copper for commercial usage.

In the rubber group, the chemists have done an excellent job in developing synthetic materials which give equally good and in some cases better service than rubber; however, the limitation of production capacity puts restrictions on a wide usage of these synthetics.

In some of the specialized parts, such as valves or similar items subjected to high temperatures, no satisfactory substitutes have been found. However, to conserve critical elements, compromises have been agreed on which reduce the percentage of scarce elements with a known reduction in the length of the service life.

At this point, let us look at a typical production vehicle such as the Chevrolet 4 x 4 Army 1½-ton truck and see how many substitutions for critical materials have been made. Our engineering records show:

107 items made of rubber have been replaced by less critical materials such as felt, cotton, fabric, plastics, synthetics, steel, iron, fiber, asphalt, paper, leather, lacquered fabric, and wood, or have been eliminated entirely by other changes.

129 items made of copper or copper-base alloys have been replaced by less critical materials such as lead, felt, steel, iron, plastics, die castings, malleable, compressed iron, granodizing, cadmium plate, zinc plate, terne plate, and cactus fiber, or have been eliminated entirely by other changes.

57 items made of tin and tin-base alloys have been replaced by less critical materials such as leather, felt, steel, iron, welding in place of soldering, granodizing, lead coating, paint, compressed iron, or tin content of the alloy reduced.

60 items made of nickel and chromium alloys have been replaced by steels with less critical elements which use very little nickel and chromium or none at all.

In each one of these substitution programs, testing and developing are still in progress to expand the number of

turn to page 93

PRODUCTION EXPERIENCE with N

IN a war in which so much emphasis is placed upon speed, research and development work have had to be thrown into high gear. Progress in the metallurgy of alloy steels has broken all speed records. Metallurgists and other scientists have had to step down from their regular laboratory procedure and perform most of their experiments on rapidly moving production lines. Most of you will recall having seen or heard, at some time or other, of the introduction of a new alloy steel. Usually after about five years of arguing, selling, and promoting, a few brave manufacturers, having pinned down their metallurgist with full responsibility, would chance a small experimental lot. The metallurgist, providing himself with insurance policies, body guards, running shoes, and other protection, would spend a year or so on the verge of a nervous breakdown until the test lot was through. If everything was satisfactory, in another five years the new steel might stand a chance of being used and written into specifications. This has not been so with the National Emergency steels. They have had the green light from the very start. They have commanded the attention of more metallurgists and engineers than any other project in the history of all industry.

Since the news was first broadcast, back in 1940, that the supply of nickel would be insufficient to meet the demands, more time and effort have been spent on research than ever before. It has been a higher type of research than we have ever experienced. We have gone deep into our old, old files and dug up information that probably never would have been used otherwise. We have had the benefit of all the latest and most up-to-date methods and tools of science.

The rapid expansion in the production of tanks, planes, guns, and ships, with its heavy demands for alloy steels, and the universal switch from nickel, brought on similar shortages in all of the alloying elements used in steel making. One by one, reports of diminishing supplies of chromium, vanadium, and molybdenum appeared. With the realization that this production of vital military equipment would have to increase, and continue to increase, not to mention civilian use and Lend-Lease, our country was faced with the problem of making more alloy steel with smaller amounts of alloying elements. To avoid costly design changes and delays, it was imperative that quality and performance be maintained.

Fortunately, all the information gained in the many years of cooperative research and the free exchange of ideas by American metallurgists has placed us at an advantage in meeting such a situation. The knowledge of the science of metallurgy has revealed to us the functions of the alloying elements in steel, singly and in combination. With this knowledge we have been able to face the problem of reducing the alloy content in our important steels and make great strides toward striking a long-range balance between production demands and available supplies.

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 13, 1943.]

It has not been an easy matter to toss aside all of our favorite alloy steels and go through the period of confusion, necessary in any major change-over. Some conservative manufacturers have hesitated to change from their 5% nickel, their chromium-nickel-molybdenum, or their chromium-vanadium steels and have been able to see nothing but trouble in making any substitutions. This is a natural attitude. It must be expected. We cannot criticize anyone for refusing to take chances on military or other vital equipment by using materials the properties of which are not known. In spite of all the theoretical knowledge we may have about alloys and alloy steels, the performance

HOW the metallurgists of industry have been forced to develop virtually a new series of National Emergency (NE) steels because the original specifications were outmoded by the huge demand on molybdenum, is explained by the author, whose services as a member of the working committee, and whose company, have been in the forefront of this metallurgical wonder of all time.

Development work which would have ordinarily taken some 10 years was crowded into a space of a few weeks. A feature of the work was digging into old files which otherwise might have gathered dust until dumped into the ashcan. The interchange of information between users and steel makers set a new high for voluntary cooperation. But the author warns that the present level of knowledge is based only upon laboratory tests and must be proceeded upon with caution.

The NE 9420 appears to be satisfactory, Mr. Roush says, as a substitute gear steel for the

of an airplane, or a tank, or a gun, in time of war, is something which must be an absolute certainty. As more experience is gained and these alloy steels are proved for their respective applications, we will see a growing willingness on the part of manufacturers to assume responsibility. Many authorities believe that these low-alloy steels, which can be made with the materials available, will become firmly established and production will continue with enough strategic alloying elements to go around.

The War Production Board has been our motorcycle escort and has set up machinery to get accurate production figures on all supplies. It can, with the information at its disposal, determine how much of any item can be used without producing a shortage, and can, by proper regulation, keep a constant flow of the proper materials to the proper places, at the proper time. On the basis of this

NATIONAL EMERGENCY STEELS

by R. W. ROUSH
Chief Metallurgist
The Timken-Detroit Axle Co.

information, the WPB suggested and promoted the development of the National Emergency steel specifications. The original allowance was ½ of 1% for nickel and chromium.

Many committees, such as the Technical Committee of the American Iron and Steel Institute, the Society of Automotive Engineers' War Engineering Board, and the industry advisory committees of the War Production Board, provided a representation of the best metallurgical engineering knowledge of the country from all viewpoints, that is, from steel makers to final fabricators.

Work was started on three basic steel compositions, namely: carbon-molybdenum, manganese-molybdenum, and chromium-nickel-molybdenum. The supply of molybdenum, at that time, was thought to be sufficient. Attempts were made to design specifications for these three series of steels, in the required carbon ranges, having an alternate in each series to match the properties of each of the alloy steels previously used.

As time would not permit a long research program or full production heats for testing, a large number of induction furnace heats were made. Adjustments were made in the carbon, manganese, and silicon contents, to balance the reductions of the principal alloying elements. Test bars were made and the standard Jominy hardenability test was used as a means of evaluation. At first, criticism was heard on the small heats of 10 to 30 lb each, and the use of hardenability tests as the only means of comparison. However, in no other way could an equal amount of information have been gathered in such a short time, as far as we know.

A great deal of emphasis is placed upon hardenability today, as a means of evaluating a steel. It is a rapid and convenient means of determining the hardening properties and a first indication of the possibilities of a steel, but to draw all of our conclusions on the basis of hardenability is neglecting the very important consideration of ductility. It is generally conceded that a steel will perform best if it has been fully hardened, all the way through, to its maximum hardness, and then tempered back to the hardness desired, and we cannot hope to get the most out of a steel unless it is so hardened. It is therefore important to know how it will harden. As tensile strength is closely proportional to hardness, it can be said that hardenability is a measure of strength, or rather the capacity for strength, especially the strength of different portions across a cross-section. The elastic limit and endurance limit are proportional to hardness, provided initial hardening was complete, but are not always consistent after incomplete initial hardening.

Elongation, reduction in area, and impact properties show a definite relationship to hardness and strength in the lower hardness ranges, but in the as-quenched condition, and in the high-hardness ranges, this relationship does not always hold, and there is a large number of parts, such as gears, which are used in these conditions. In this case, laboratory or performance tests on actual parts are the only safe and sure means of proving the value of a steel. The hardenability test has been a wonderful yardstick in building the NE steel specifications, and was probably the only one suitable, but for the above reasons the real value of a steel should be confirmed and verified by laboratory and performance tests. This is being done as rapidly as possible, and steels not satisfactory are being either altered or deleted.

There are applications which do not require full, deep hardening; in fact, some parts perform better if they do not harden throughout. In shafts and members subjected to torsional stress only, it has been found that a high surface hardness and a lower center hardness give the best torsional fatigue properties. Then there are a number of automotive parts being used today which have never been fully hardened and yet are giving good performance. Their design is adequate, and full hardness and strength are not required.

Hardenability may be determined in a number of different ways. The most common is the Jominy or end quench method. Briefly, it consists of cooling the end of

former NE 4120, and may probably be used as a substitute for the NE 4620 specification. However, it does not appear to be warranted as a substitute for the NE 4320 or NE 4820. For these, he reported, the NE 8720 does appear to be satisfactory.

A great deal more work will have to be done before designers can be sure of the newest substitutes for substitute steels, he concludes, but counts this effort to save critical materials a factor in winning the war.

■ ■ ■

THE AUTHOR: R. W. ROUSH, chief metallurgist for The Timken-Detroit Axle Co., has served on various committees for the selection and application of alternate steels. Among the committees on which he has served are the Automotive Equipment Subcommittee of the Technical Advisory Committee on Carbon and Alloy Steel Bars, Billets, Blooms and Slabs, War Production Board; the Iron and Steel Subcommittee of the SAE War Engineering Board, and various subcommittees of the AISI's Technical Committee on Alloy Steels.

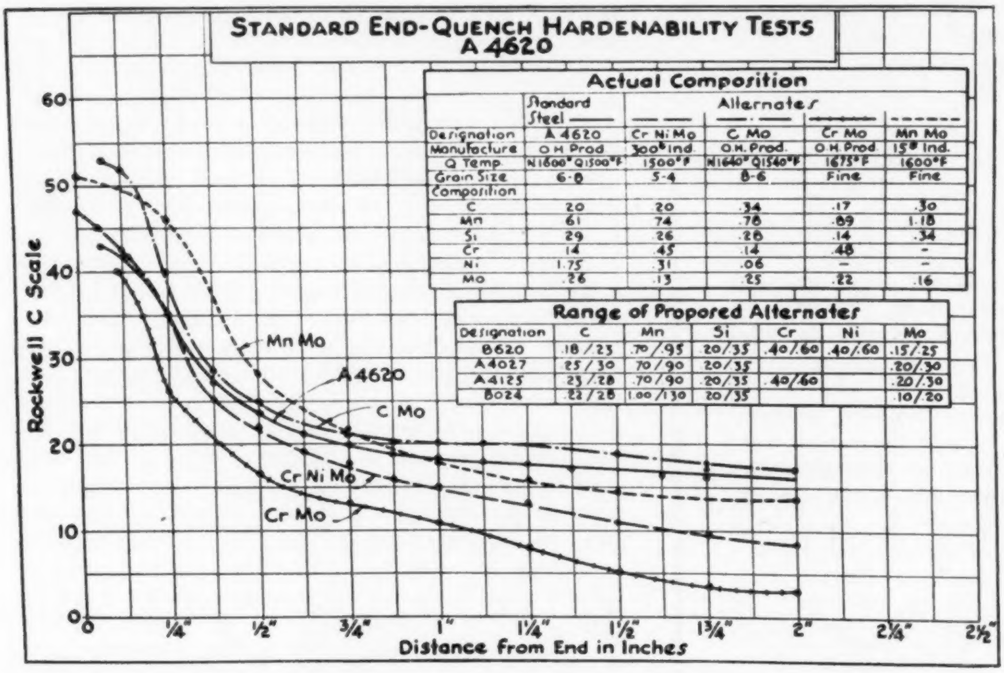
a 1-in. round, 3-in. long bar, in a stream of water of a constant temperature and pressure, and measuring the Rockwell hardness at 1/16-in. intervals along the side of the bar beginning at the water-cooled end[1]. Hardness is plotted against distance from the water-cooled end, showing the familiar hardenability curve. By correlating these results with hardness tests made on various sized parts, a close prediction can be made of the hardness possible in a desired section. This is a standard method and is given in full in the "SAE Handbook."

Another method is the through hardness method, which consists of taking the Rockwell hardness at 1/16-in. intervals across the cross-section of a round bar or a finished part after it has been properly quenched, and plotting a curve showing hardness at all points across the section. This is useful and a direct method of observing the hardness of shafts and round sections, where a definite surface and center hardness is desired. While it is possible to get some variation in hardenability from steels of the same type, the complete chemical composition and the grain size offer a means of predicting very closely the hardenability. Using the hardenability factors for the alloying elements shown by Grossman[2], it is possible to produce hardenability in many different ways with the alloys available.

Fig. 1 shows the hardenability curve for a popular gear steel, SAE 4620. With it are curves for steels of three basic

[1] See ASM Transactions, Vol. 26, 1938, pp. 574-599: "A Hardenability Test for Carburizing Steel," by W. E. Jominy and A. L. Boegehold.

[2] See AIME Technical Publication No. 1437 (Metals Technology), Feb., 1942: "Hardenability Calculated from Chemical Composition," by M. A. Grossman.



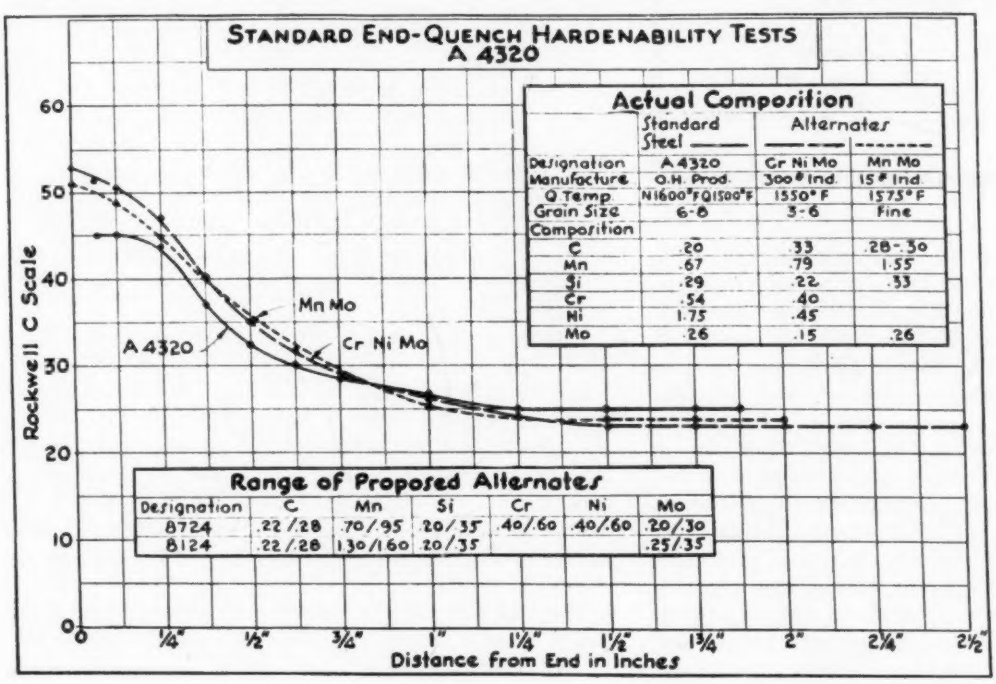
Actual Composition

	Standard Steel	Alternates			
Designation	A 4620	Cr Ni Mo	C Mo	Cr Mo	Mn Mo
Manufacture	O H Prod	300# Ind.	O.H. Prod.	O H Prod	15# Ind.
Q Temp	N1600° Q1500°F	1500°F	N1640° Q1540°F	1675°F	1600°F
Grain Size	6-8	5-4	8-6	Fine	Fine
Composition					
C	20	20	.34	.17	.30
Mn	61	74	.78	.89	1.18
Si	29	.26	.28	.14	.34
Cr	14	.45	.14	.48	-
Ni	1.75	.31	.06	–	-
Mo	.26	.13	.25	.22	.16

Range of Proposed Alternates

Designation	C	Mn	Si	Cr	Ni	Mo
8620	.18/.23	.70/.95	.20/.35	.40/.60	.40/.60	.15/.25
A4027	25/30	70/90	20/35			.20/.30
A4125	23/.28	.70/90	.20/.35	.40/.60		.20/.30
8024	.22/28	1.00/130	20/35			.10/.20

■ Fig. 1 – Hardenability curves for SAE 4620 and alternates



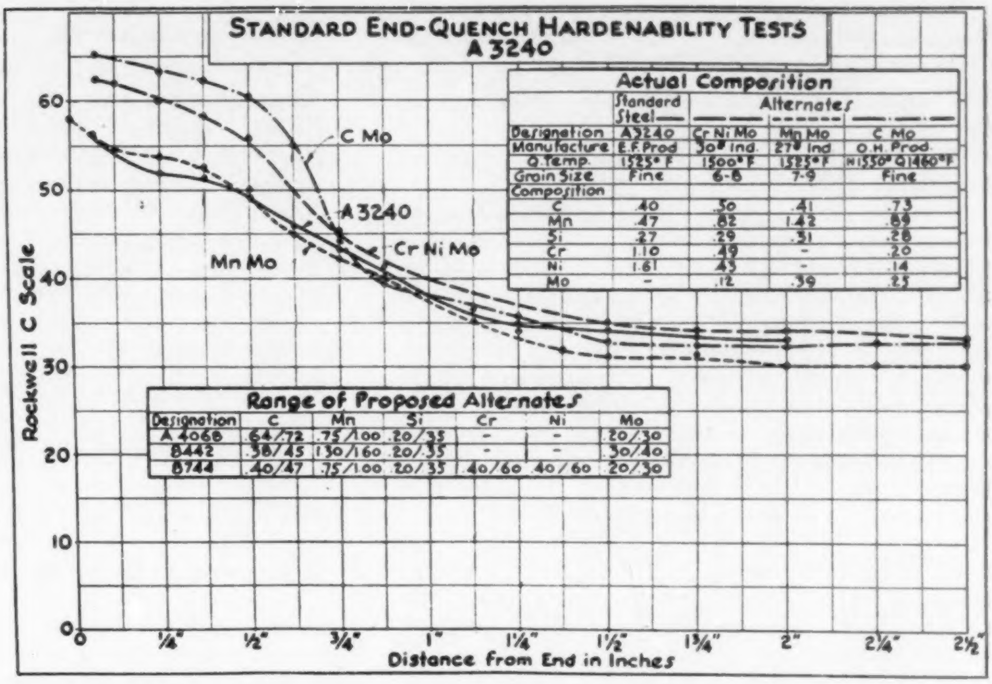
Actual Composition

	Standard Steel	Alternates	
Designation	A 4320	Cr Ni Mo	Mn Mo
Manufacture	O.H. Prod.	300# Ind.	15# Ind.
Q Temp.	N1600°F Q1500°F	1550°F	1575°F
Grain Size	6-8	3-6	Fine
Composition			
C	.20	.33	.28-.30
Mn	.67	.79	1.55
Si	.29	.22	.33
Cr	.54	.40	
Ni	1.75	.45	
Mo	.26	.15	.26

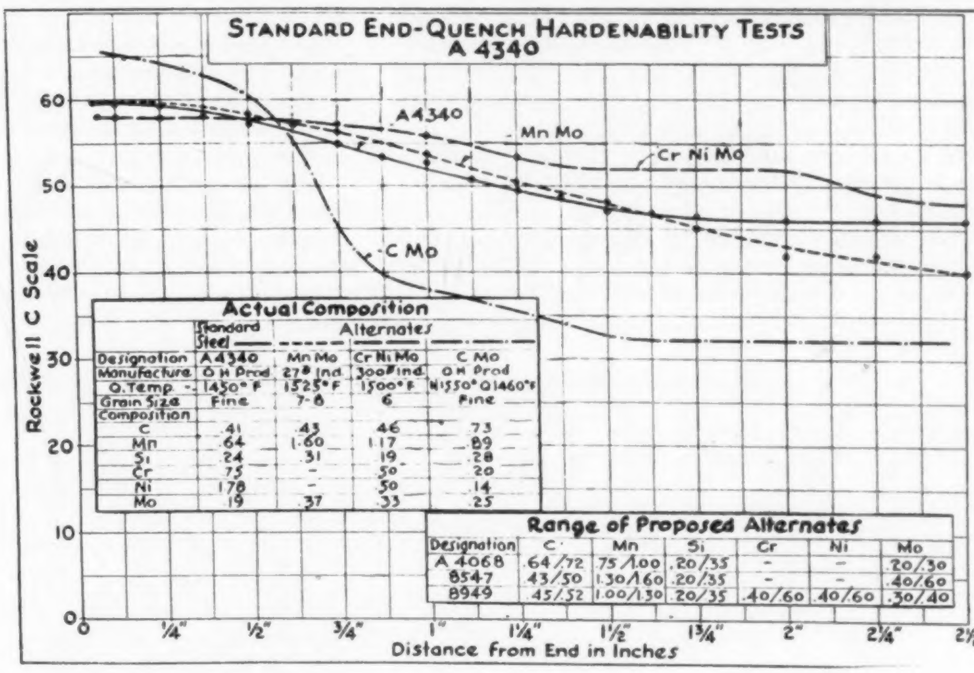
Range of Proposed Alternates

Designation	C	Mn	Si	Cr	Ni	Mo
8724	.22/.28	.70/.95	.20/.35	.40/.60	.40/.60	.20/.30
8124	.22/.28	1.30/1.60	.20/.35			.25/.35

■ Fig. 2 – Hardenability curves for SAE 4320 and alternates



■ Fig. 3 – Hardenability curves for SAE 3240 and alternates



■ Fig. 4 – Hardenability curves for SAE 4340 and alternates

compositions using much smaller quantities of the alloying elements, nickel and chromium. The hardenability curves are very similar and are well within normal allowable limits. Aside from the manganese-molybdenum and the chromium-molybdenum, the curves are very comparable.

In Fig. 2 the hardenability curve for SAE 4320 has been almost exactly duplicated with a manganese-molybdenum and a chromium-nickel-molybdenum steel. This shows that these steels have a similar hardness in a given section. For instance, if we have been getting a certain hardness in the center at the root of a gear tooth when using SAE 4320, we know we can get the same hardness at the same point in the same gear, made of this manganese-molybdenum, or this chromium-nickel-molybdenum steel. This manganese-molybdenum steel was designated as 8124 and the chromium-nickel-molybdenum as 8724. The 8124 has since been deleted and the 8724 altered to 8720.

A structural grade is shown in Fig. 3, SAE 3240. The steels proposed to match this hardenability curve were:

Type	Designation
Carbon-Molybdenum	A 4068 (now deleted)
Manganese-Molybdenum	NE 8442
Chromium-Nickel-Molybdenum	NE 8744

These curves are close enough to give assurance that an equal hardness can be obtained in a given section in either of these steels, even though they have different compositions.

Fig. 4 shows curves for deep-hardening grades. The hardenability curve for SAE 4340 has been matched by:

Type	Designation
Manganese-Molybdenum	NE 8547 (now deleted)
Chromium-Nickel-Molybdenum	NE 8949

In this case the carbon-molybdenum (4068) does not have the hardenability.

These deep-hardening grades are used for heavy sections where other steels will not harden through.

All of the common alloy steels were matched in this manner with alternate steels having similar hardenability.

With the commercial limits necessary to allow in any steel, it is natural that we cannot expect to match the hardenability exactly to the line. In the core hardness at any point in a gear, for instance, we have to have limits. We have determined by performance testing that a given gear will be satisfactory with a core hardness range of, say, 10 points, Rockwell C.

In specifications for hardenability we likewise have to consider limits. We therefore establish a hardenability band instead of a single-line curve.

One example of this is shown in Fig. 5, which is the hardenability band showing the limits we require and obtained with 30, 100-ton production heats of 8744 for axle drive shafts. The curve outside the band represents a heat which would not give the required hardness in production.

The first specifications for the NE steels to be published were as shown in Fig. 6. This covers the manganese-molybdenum, 8000 to 8500 series, and the chromium-nickel-molybdenum types, 8600 to 8900, inclusive. The carbon-molybdenum steels were designated as the 4000 series, for example 4023 and 4042. Alternates for practically all of the alloy steels previously used were selected from this list.

These steels were ordered by manufacturers, and tests made in production and in the laboratory. Some trouble was experienced with the manganese-molybdenum series in the low-carbon carburizing grades. Core hardnesses were high, and ductility and impact strength were low. Improved heat treating practice made a great difference and good gears have been produced from the 8024 and 8124.

Des.	C	Mn	Mo	Ni	Cr
NE 8024	0.22/0.28	1.00/1.30	0.10/0.20	—	—
NE 8124	0.22/0.28	1.30/1.60	0.25/0.35	—	—
NE 8233	0.30/0.36	1.30/1.60	0.10/0.20	—	—
NE 8245	0.42/0.49	1.30/1.60	0.10/0.20	—	—
NE 8339	0.35/0.42	1.30/1.60	0.20/0.30	—	—
NE 8442	0.38/0.45	1.30/1.60	0.30/0.40	—	—
NE 8447	0.43/0.50	1.30/1.60	0.30/0.40	—	—
NE 8547	0.43/0.50	1.30/1.60	0.40/0.60	—	—
NE 8620	0.18/0.23	0.70/0.95	0.15/0.25	0.40/0.60	0.40/0.60
NE 8630	0.27/0.33	0.70/0.95	0.15/0.25	0.40/0.60	0.40/0.60
NE 8724	0.22/0.28	0.70/0.95	0.20/0.30	0.40/0.60	0.40/0.60
NE 8739	0.35/0.42	0.75/1.00	0.20/0.30	0.40/0.60	0.40/0.60
NE 8744	0.40/0.47	0.75/1.00	0.20/0.30	0.40/0.60	0.40/0.60
NE 8749	0.45/0.52	0.75/1.00	0.20/0.30	0.40/0.60	0.40/0.60
NE 8817	0.15/0.20	0.70/0.95	0.30/0.40	0.40/0.60	0.40/0.60
NE 8949	0.45/0.52	1.00/1.30	0.30/0.40	0.40/0.60	0.40/0.60

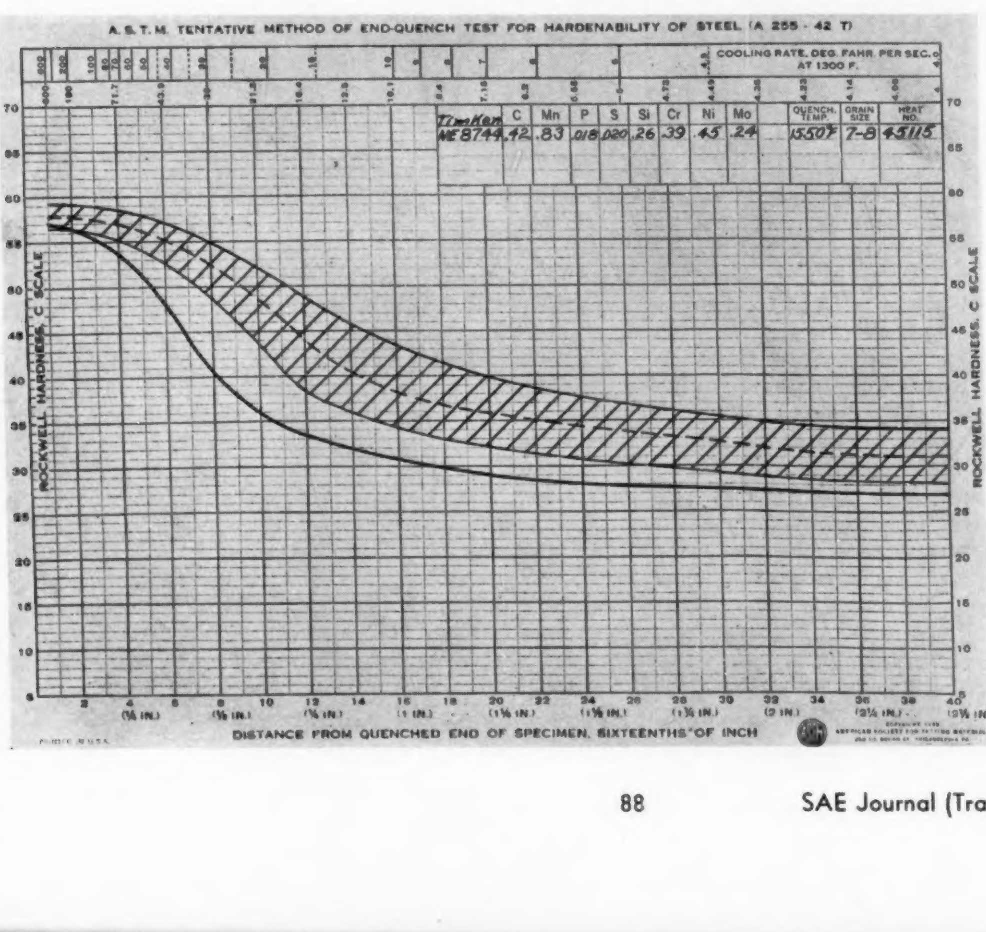
NOTE. The following additional element specifications apply to all the basic open hearth steel shown above: Phosphorous 0.040% max.; Sulphur 0.040% max.; Silicon 0.20, 0.35%.

■ Fig. 6 – Original NE steel compositions

The 8442 is an excellent steel and has been used for a number of structural parts. It has been used for studs, bolts, capscrews (both machined and cold headed), and when hardened to between 200 and 400 Brinell it shows excellent properties.

Since greater shortages have occurred in molybdenum these steels have been practically eliminated.

By far the most popular of the NE steels have been the 8600 series and 8700 series. In the carburizing grades, either makes good gears. While there is only five points difference in molybdenum, the 8700 series is probably



■ Fig. 5 – Hardenability band for NE 8744

better for heavy-duty gears. The 8630 is used for a number of structural parts. It can be water quenched, and with between 200 and 400 Brinell shows good fatigue properties.

About the time the steels of the 8000 series were being well established and their use for important applications showing a gradual growth, further shortage of the strategic alloying elements, brought about by increased alloy steel production, made necessary further reductions in order to ensure uninterrupted production of alloy steels. Shortages in molybdenum have become very serious, making it necessary to conserve on this alloy wherever possible. Watchful and careful checking of the scrap situation showed good possibilities for a chromium-nickel-molybdenum series of steels, which could be made with little or no alloy additions. The alloy contents were:

Chromium	0.20-0.40%
Nickel	0.20-0.40%
Molybdenum	0.08-0.15%

By making experimental heats with these alloy contents, plus slight modifications in silicon and manganese, hardenabilities were found to compare favorably with those of all of the important alloy steels. This has led to the formation of a supplementary list of National Emergency steel specifications. The first of these is the 9400 series with the very low alloy content. The next is the 9500 series, which is the same as the 8600 series with additional silicon and manganese. This series has high hardenability and is intended for heavy sections. As the chromium situation looked a little better the 9600 series has been added. These steels have chromium, silicon, and manganese as the principal alloying elements.

Within the last month it has been necessary to alter some of these specifications on account of the alloy content of the available scrap. The percentage of nickel has been running higher than was anticipated and the limits for this element are being broadened to allow a wider range. Strange as it may seem, the percentage of nickel in the 8600 and 8700 series will be changed from 0.40-0.60 to 0.40-0.70, and in the 9400 series from 0.20-0.40 to 0.20-0.50.

The extremely critical condition in molybdenum has forced greater reductions in this element, and the 8700 series, except for the 8720, is being deleted. The 8817 and 8949, and also the 8339, 8442, and 8547, having high molybdenum contents, have all been deleted.

The latest list is shown in Fig. 7.

In eliminating the 8700 series and substituting the 8600 we are sacrificing 0.05% molybdenum. The effect of this can be seen in Figs. 8 and 9, which show hardenabilities of a 9420 and a 9430 analysis with 0.15% molybdenum, 0.08% molybdenum, and no molybdenum.

From actual production experience with the NE steels, there have been a few cases reported of flakes or bursts in large forging bars. This happens in the larger sizes and is especially prevalent in the steels of high hardenability. The condition is remedied by slow cooling after rolling. The mills have learned that the same care and slow cooling required for the SAE alloy steels are required for the NE steels. The writer encountered this only in one heat of 8720. This heat had to be scrapped as it cracked in cold shearing and could not be used for forgings. The NE steels are no worse in this respect than any others.

In forging there has been no very great difference between the old SAE alloy steels and any of the NE steels.

As it has been necessary for all manufacturers to use

CARBON-MANGANESE STEELS

	C	Mn	Si
NE 1330	0.28/0.33	1.60/1.90	0.20/0.35
NE 1335	0.33/0.38	1.60/1.90	0.20/0.35
NE 1340	0.38/0.43	1.60/1.90	0.20/0.35
NE 1345	0.43/0.48	1.60/1.90	0.20/0.35
NE 1350	0.48/0.53	1.60/1.90	0.20/0.35

MANGANESE-MOLYBDENUM STEELS

	C	Mn	Si	Mo
NE 8020	0.18/0.23	1.00/1.30	0.20/0.35	0.10/0.20
NE 8442*	0.40/0.45	1.30/1.60	0.20/0.35	0.30/0.40

NICKEL-CHROMIUM-MOLYBDENUM STEELS

	C	Mn	Si	Cr	Ni	Mo
NE 8613	0.12/0.17	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8615	0.13/0.18	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8617	0.15/0.20	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8620	0.18/0.23	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8630	0.28/0.33	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8635	0.33/0.38	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8637	0.35/0.40	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8640	0.38/0.43	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8642	0.40/0.45	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8645	0.43/0.48	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8650	0.48/0.53	0.75/1.00	0.20/0.35	0.40/0.60	0.40/0.70	0.15/0.25
NE 8720	0.18/0.23	0.70/0.90	0.20/0.35	0.40/0.60	0.40/0.70	0.20/0.30

SILICON-MANGANESE AND SILICON-MANGANESE-CHROMIUM STEELS

	C	Mn	Si	Cr
NE 9255	0.50/0.60	0.70/0.95	1.80/2.20	—
NE 9260	0.55/0.65	0.75/1.00	1.80/2.20	—
NE 9262	0.55/0.65	0.75/1.00	1.80/2.20	0.20/0.40

MANGANESE-SILICON-CHROMIUM-NICKEL-MOLYBDENUM STEELS

	C	Mn	Si	Cr	Ni	Mo
NE 9415	0.13/0.18	0.80/1.10	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9420	0.18/0.23	0.80/1.10	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9422	0.20/0.25	0.80/1.10	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9430	0.28/0.33	0.90/1.20	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9435	0.33/0.38	0.90/1.20	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9437	0.35/0.40	0.90/1.20	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9440	0.38/0.43	0.90/1.20	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9442	0.40/0.45	1.00/1.30	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9445	0.43/0.48	1.00/1.30	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9450	0.48/0.53	1.20/1.50	0.40/0.60	0.20/0.40	0.20/0.50	0.08/0.15
NE 9537*	0.35/0.40	1.20/1.50	0.40/0.60	0.40/0.60	0.40/0.70	0.15/0.25
NE 9540*	0.38/0.43	1.20/1.50	0.40/0.60	0.40/0.60	0.40/0.70	0.15/0.25
NE 9542*	0.40/0.45	1.20/1.50	0.40/0.60	0.40/0.60	0.40/0.70	0.15/0.25
NE 9550*	0.48/0.53	1.20/1.50	0.40/0.60	0.40/0.60	0.40/0.70	0.15/0.25

MANGANESE-SILICON-CHROMIUM STEELS

	C	Mn	Si	Cr
NE 9630	0.28/0.33	1.20/1.50	0.40/0.60	0.40/0.60
NE 9635	0.33/0.38	1.20/1.50	0.40/0.60	0.40/0.60
NE 9637	0.35/0.40	1.20/1.50	0.40/0.60	0.40/0.60
NE 9640	0.38/0.43	1.20/1.50	0.40/0.60	0.40/0.60
NE 9642	0.40/0.45	1.30/1.60	0.40/0.60	0.40/0.60
NE 9645	0.43/0.48	1.30/1.60	0.40/0.60	0.40/0.60
NE 9650	0.48/0.53	1.30/1.60	0.40/0.60	0.40/0.60

CARBON-CHROMIUM STEELS

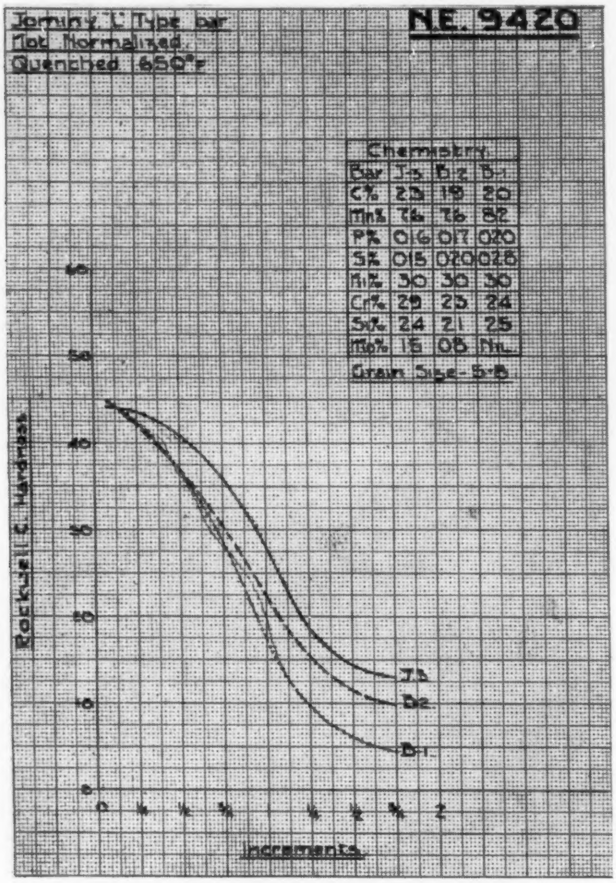
	C	Mn	Si	Cr	Ni	Mo
NE 52100A	0.95/1.10	0.25/0.45	0.20/0.35	1.30/1.60	0.35 max.	0.08 max.
NE 52100B	0.95/1.10	0.25/0.45	0.20/0.35	0.90/1.15	0.35 max.	0.08 max.
NE 52100C	0.95/1.10	0.25/0.45	0.20/0.35	0.40/0.60	0.35 max.	0.08 max.

* Recommended for large sections only.

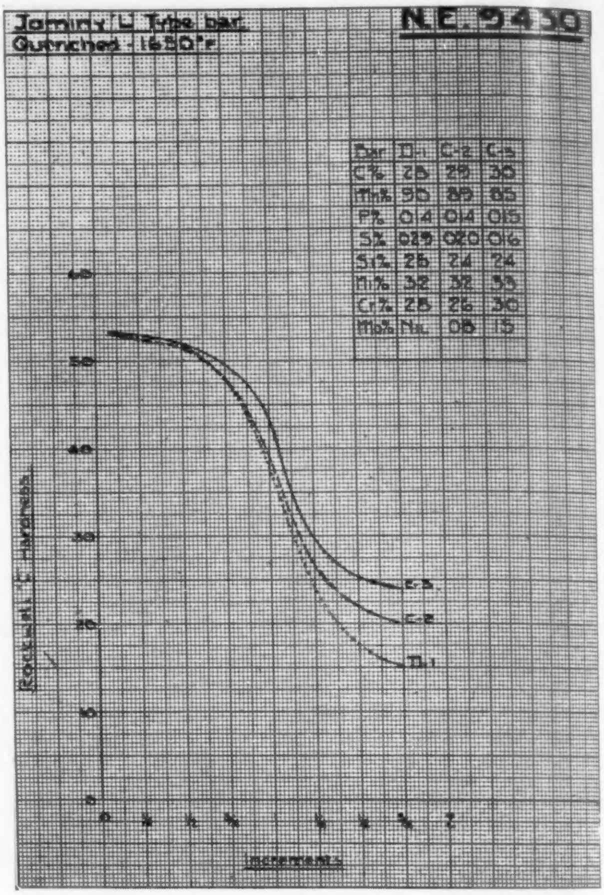
■ Fig. 7 – Revised NE steel compositions – AISI, section 10, sheet 182, published Dec. 17, 1942

their available equipment for normalizing and annealing, there has been no great change in cycles in changing to the NE steels. Almost all of the steels respond to the cycles formerly used for the SAE alloy steels. The carburizing grades of NE steels apparently require less time on cooling than some of the old SAE. Some special short annealing cycles have been worked out, and there will undoubtedly be more, which will show economies in time, fuel, and machining.

A number of manufacturers have reported no difference in machining between the NE steels and the former SAE steels. In some cases large lots have gone through production lines without the operators knowing there was a change in material. As more NE steels are used, more complete records are kept and a better overall picture is given of machining properties. In our plant we have made approximately 100,000 spiral bevel rear-axle drive gear sets from NE 8720 steel. This involves some 20 heats of 100 tons each. In the rough and finish cutting of these gear teeth, complete records are kept of the number of gears per cutter grind. Results, in comparison to SAE 4620 and 4120, are shown in Table 1.



■ Fig. 8 – Hardenability curves for NE 9420



■ Fig. 9 – Hardenability curves for NE 9430

Table 1 – Machining Spiral Bevel Gears

Generating 1¾-in. Tooth – 28 sec
(1.2 oz stock removed per tooth)

Steel	Gears per Cutter Grind 4620	4120	8720
Rough, 178 fpm	90	90	90
Finish, 157 fpm (0.008 in. cut)	50	40	40
Formate 1¾ in. Tooth – 8 sec			
Rough, 178 fpm	200	190	190
Finish, 157 fpm	600	400	400

Brinell 162

Turning is done at 178 fpm, using a 0.010-in. feed and a 5/32-in. depth of cut. Brinell hardness is from 152 to 167, with an average of 162.

Fig. 10 shows a 13-in. spiral bevel assembly with nine important parts, and Fig. 11 shows a two-speed double reduction with 15 important parts of NE steel carburized and hardened.

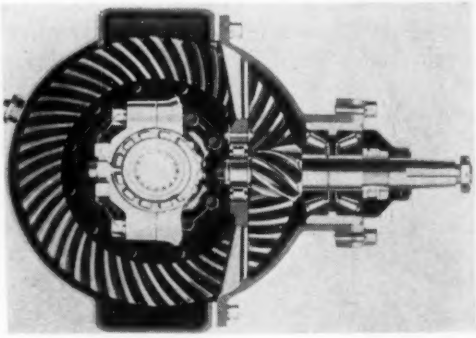
In machining NE 8744, compared to SAE 3240 and 4340 at high hardness, our records for a large production are shown in Table 2.

Table 2 – Hobbing of Splines at 400 to 444 Brinell

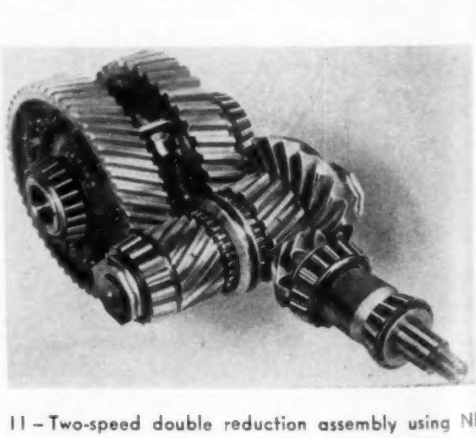
Cutter speed, 72 rpm — Surface fpm, 84 — Depth of cut, 5/32 in. Feed 0.0057 in.

Steel	3240	4340	8744
Pcs. per grind	46	46	34
Pcs. per cutter	785	785	578

NE 8949 could not be machined with any success at this high hardness. At 375 to 430 Brinell it can be machined, but cutter life drops 80% from that of 4340.



■ Fig. 10 – Spiral bevel assembly using NE steels



■ Fig. 11 – Two-speed double reduction assembly using NE steels

In the normalized condition, these two steels, NE 8744 and 8949, machine very favorably in comparison with SAE 3240 and 4340. Cutting speeds range from 178 fpm with high-speed tools to 460 fpm with carbide tools.

At hardnesses of 200 to 350 Brinell, machining of the NE steels is comparable to the corresponding SAE grades.

8630 is comparable to 3130 and 4130
8735 to 8740 are comparable to 3135 and 4140
8740 to 8745 are comparable to 3240 and 4340

A considerable amount of work has been done on carburizing of the NE steels. Chemical analyses for carbon content of the case, carburized in gas and solid compound, have been made and published in a number of recent papers[3]. In general, the NE steels, especially the more popular grades 8620 and 8720, absorb slightly more carbon and show a higher carbon concentration on the surface. Carbon content, microstructure, and hardness at various depths agree closely with the SAE grades, 4120, 4320, and 4620.

Some manufacturers are reluctant to use an NE steel as a substitute for some of the so-called high-powered steels such as 2512 and 3312. We started out by substituting 8817 for these steels and from all indications it gave promise of good performance. Since it has been deleted, attempts are being made to substitute NE 8720. No doubt it will suffice for some applications, but whether or not it will meet all the requirements of the 2512 and 3312 remains to be seen. Some users have expressed the opinion that there should be an 8920. At least we can say that good judgment was used in retaining the 8720 when all of the other 8700 steels were deleted.

Opinions are divided as to substituting NE 8720 for SAE 4820, although this is being worked successfully in a number of applications.

Parts made from 8620 and 8720 have been quenched direct from the carburizing temperature, cooled slowly, and reheated for hardening, and also double quenched. All treatments have been successful in their respective places. It has been found that a higher reheating temperature is required than was used for the same parts made of the SAE steels. It is our belief that these steels are at their best when direct quenched, especially for many heavy-duty applications.

Distortion in hardening of these steels is comparable to 4620. Spiral bevel gears in our plant run 1½ to 2% rejections for out-of-round and out-of-flat.

[3] See *Steel*, Vol. 111, No. 24, Dec. 14, 1942, pp. 99, 119-123: "A Metallurgical Study of Some NE Alloy Steels," by Thomas A. Frischman.

Carbon determinations of the case on 9420, carburized in gas and in compound, show it to be equivalent to 4620 and 4320 in this respect. Some manufacturers have reported the 9420 acceptable for 4120, but there has been much hesitancy in substituting it for 8620, 8720, or 4620. A tremendous amount of work is in progress on this steel at the present time.

For structural steels, both oil and water hardening of a large tonnage of the 8600 and 8700 series have been used. In the manganese-molybdenum grades, NE 8339 and 8442, although now deleted from standard lists, have been used extensively. Also, the use of NE 9442 for a number of parts is increasing. In the Brinell hardness range of 200 to 400, the available physical properties show that all of these steels are satisfactory. They have good strength and toughness.

Fig. 12 shows physical properties for NE 9442.

For studs, bolts, and capscrews no trouble should be experienced.

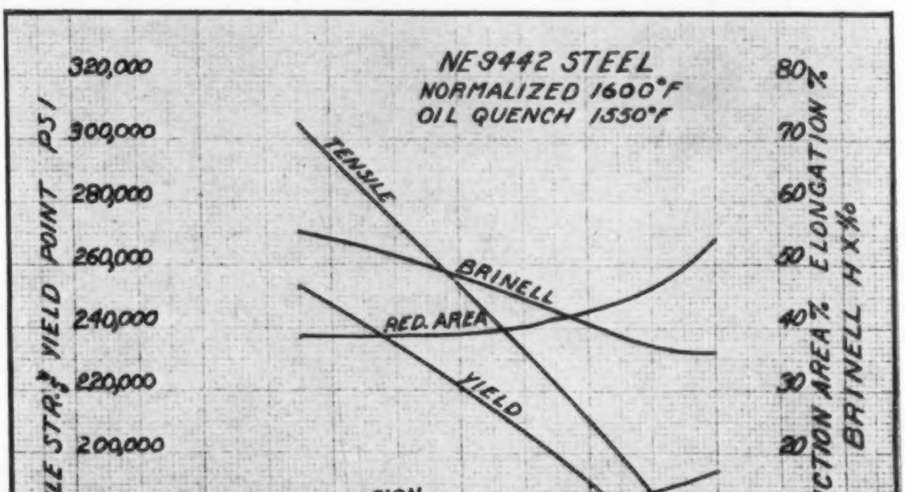
NE 8630 has been substituted successfully for 3130 and 4130. There has been considerable production experience on this steel for steering arms and knuckles; and a limited number of fatigue tests at 300 Brinell hardness show that it is equivalent to the original steels.

NE 8744 has been used extensively for axle drive shafts in diameters up to 2¼ in. They are quenched from 1550 F into oil and give a hardness of 500 to 550 Brinell. Tempering at 800 F gives 400 to 444 Brinell.

Quantities of shafts have been made of 8949. They have deep-hardening qualities and show reverse torsional endurance limits in excess of those for 3240 and 4340.

Some trouble has been experienced in cracking of parts made of 8949. It is especially adapted to heavy sections and should not be used in parts where an 8700 steel will meet the hardenability requirements.

Considerable work has been done on different types of welding on a limited number of the NE steels. NE 8630



■ Fig. 12 – Physical properties of NE 9442

seems to be one of the best suited. It has replaced 4130 in a number of applications. Aircraft tubing has been welded very successfully, both by oxyacetylene gas flame and metallic arc. The same general technique can be used for either steel. Using a mild steel rod, the maximum hardness reached in the heat-affected zone is approximately the same for both steels. The warping and shrinking characteristics are similar, likewise the susceptibility to cracking. The tensile strengths of welded joints, both as welded and heat treated, are equivalent for both steels. Both fail in the same manner. While the 8630 is the only NE steel on which an appreciable amount of welding information is available, it is estimated that 9430 will show comparable results.

No discussion dealing with steels, poor in alloy, could be complete without considering the effects of special alloy addition agents (ladle additives). You have all heard the various names given to these steels, such as super-duper, suped up, needled, vitalized, vitaminized, intensified, a shot in the arm, Irish stew, and many others. Regardless of what we may think or say, these steels are going to be in the picture because they do give results. With their high ductility at high hardness they have a combination of properties which is not equaled by any other alloy steels. Without attempting to explain the exact mechanism of their action, I just want to report the results of a few tests made in our laboratory and some from other laboratories. After about four years of experience, with vanadium-type additive treatment in the carburizing and structural grades of steel showing amazing results, we could not sit back and think of it all as just a memory. Since vanadium has been restricted we have continued the use of these specially treated steels, and have tried many different types. We believe today that a proper alloy addition agent will give to the 9400 series the qualities which will make it a high-grade steel.

We have recently had one heat of a modified 9420 with an additive treatment which, when made into spiral bevel rear-axle drive gears, passed our maximum requirements on the dynamometer test.

A very good report was received from a large automotive manufacturer on dynamometer testing of carburized and hardened transmission gears. The following information was given on the gears: diametral pitch, 5; pressure angle, 20 deg; tooth width, 9/16 in.; ratio, 15 to 25; chamfer, none. Gears were carburized, direct quenched, and tempered at 380 F to 400 F; shot blasted; tested at 1200 rpm pinion speed. Results are shown in Table 3.

Table 3 – No. of Cycles to Cause Failure at 100,000 psi Bending Stress

NE 9420	Grainal	819,000
NE 8620		472,900
NE 9420		204,100
NE 8124		109,100

These tests were made on a specially designed dynamometer to test the gears only and not the complete assembly.

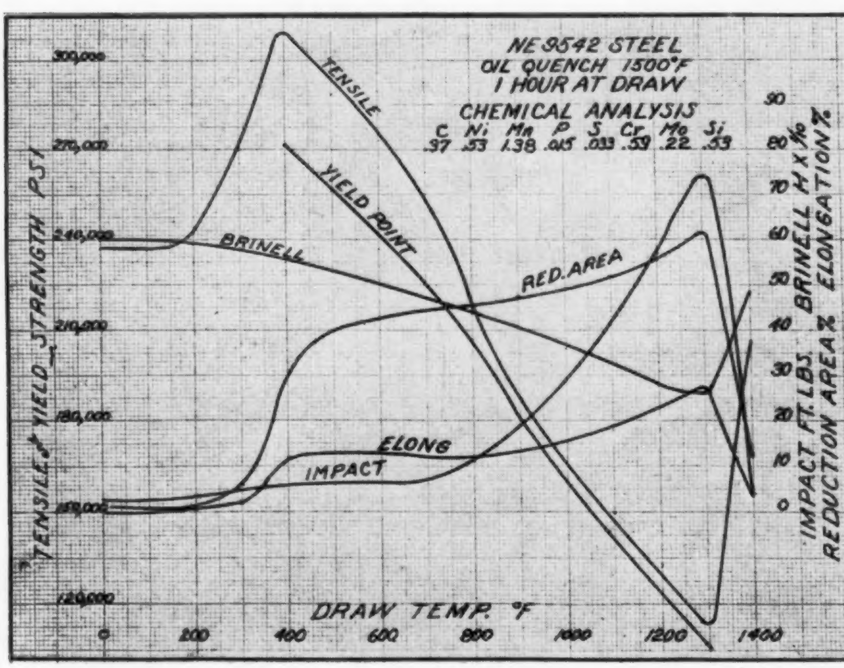
When performance like this is shown repeatedly it is something we cannot overlook. Very capable committees are at work and programs are in progress for extensive tests. Now that we have the go signal for these special alloy addition agents we can expect greater developments.

In summarizing the data available on the NE steels, we must proceed with caution. The number of tests reported to date is not large and the results cannot be taken as too conclusive. While it is all preliminary and subject to change, a start must be made sometime, so we will be helping the war effort by using the NE steels. It is our duty to make tests and improvements and apply these steels wherever possible, and conserve strategic alloy.

In the gear steels the 9420 is satisfactory as a substitute for 4120 and probably for 4620. It is not quite the equivalent of 4320 or 4820. For most applications reported, the 8720 has been satisfactory for these steels. In the water-hardening structural grades, the 8630 can be substituted for 3130 and 4120. Judging from physical properties, it is reasonable to believe that 9430 will approach the 8630. The 9442 shows physical properties, including torsion, comparable to 4140.

The 8739 to 8749 have been very good substitutes for 3135 and 4140, and their being deleted in favor of the 8600 series, meaning only a reduction of 0.05% molybdenum, should not make a great deal of difference. But when alloys are reduced to the minimum, five points of moly are extremely important. We believe it is good judgment to retain the 8720 for gears and a number of carburized parts.

From the very small amount of data available on the 9600 series, it appears to lack ductility. Fig. 13 shows physical properties for NE

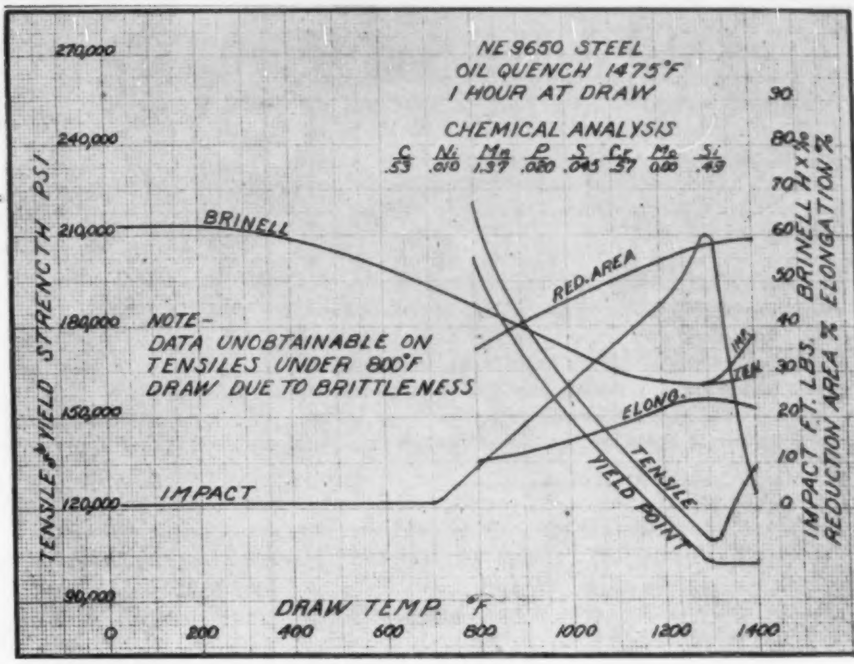


■ Fig. 13 – Physical properties of NE 9542

9512, and Fig. 14 physicals for NE 9650. If it is used, it should be hardened between 200 and 400 Brinell. Careful and improved heat-treating practice will have to make up for some of the weaknesses.

Some of you wonder why there have been so many changes in NE specifications; however, it has been a question of making alloy steel with the alloys we have, and alloy steels are still being made. There is a lot of work still to be done on the NE steels, especially some of the last to be introduced; also on the steels treated with addition agents. With a war to win, it is our duty to see that this work is done and that our military equipment remains the best.

Appreciation for valuable assistance in preparing this discussion is extended to E. O. Mann, Chevrolet Motor Division, General Motors Corp., W. P. Eddy, General Motors Truck Co., H. B. Knowlton, International Harvester Co., W. E. Day, Jr., Mack Manufacturing Corp., and to all of the members of the SAE War Engineering Board, Iron & Steel Committee.



■ Fig. 14 – Physical properties of NE 9650

Substitute Materials – Have We Gone the Limit?

continued from page 83

substitutions already in effect. Many of these are nearly ready to be released for production. The progress just described applies to only one vehicle taken as a typical example. There are other vehicles that have received the same treatment; none have been overlooked.

Have we reached the limit of substitutions? Considered from the standpoint of war materials, probably not – as it stands to reason that if we are forced to use lower grade materials than those now being used, then ways and means *will* be found to do this, following along the lines laid down in the early part of this paper. If, in the case of steels, we reach the point where no alloys are available, then a lot of redesigning of parts will be necessary and interchangeability may be sacrificed in some instances.

By this we mean that if it becomes necessary to make an Army vehicle of cast iron, then we will be forced to design one that will permit the use of such material. It will be heavier, parts will be much larger to withstand the stresses and loads, and durability will be very much inferior, but we will still make a vehicle and it will be made to do the job required of it. The extent to which we will be forced to go cannot be exactly predicted as it depends entirely upon the length of the war and the availability of materials for each particular vehicle or war unit.

However, we are certain of one thing: our engineering resources are not exhausted nor do we intend to allow any reasonable demand to exhaust those resources! This country has the best engineering talent in the world and it challenges any demand upon it for a showdown! We are not considering ourselves supermen, but we *can* produce implements of war that will more than match the enemy's and we may have to do it without the use of restricted alloys. We may be forced to do it with the lowest grade of iron, and if we are, you can be assured that, if at all possible, *it will be done!*

Therefore, in answer to the question, "Have we gone the limit of substitutions?" we definitely say, "No!" Our progress from this point on, and how far we must go, will depend entirely upon the supply of materials.

If we attempt to look into the postwar possibilities of substitute materials, the conclusions are that economics will be the principal factor to determine the situation. For example, when nickel and chromium are no longer restricted, it is more likely that such elements will be used extensively in steels to replace lower grade alloys – and while it may be that the cost of the basic raw material is higher in the case of the better alloys than in the case of the lower alloys, the heat-treat range is greater and the handling cost in processing is thus lower, and the overall cost is less.

In the case of aluminum, there is a general belief that its cost will be much lower after the war because of the widely expanded production facilities, and if this becomes a reality, no doubt aluminum will be more extensively used.

As to the outcome of the position of rubber versus synthetics, this will most certainly be determined by the relative cost of the two materials, as in most instances there is no particular advantage of one over the other.

The automotive industry has always been alert to any economic advantages to be gained through materials or methods, so that whatever may be learned by the war effort, if it is worth while, will not be lost after the war.

A NUMBER of methods for supplying heat to the incoming charge of air sufficient to obtain combustion under subzero temperatures are described by Mr. Pelizzoni. This extra heat, he explains, must be supplied because under initial cranking conditions, the heat of compression does not provide a temperature sufficiently high for self-ignition of the fuel.

Mr. Pelizzoni divides the methods of heating the incoming air into two general types: preheaters, which are used with the engine still motionless, and cranking heaters, which have the heat applied while the engine is being cranked. The necessary heat may be obtained either electrically or by the combustion of some of the diesel fuel itself.

THE AUTHOR: W. J. PELIZZONI (M '40) joined the International-Plainfield Motor Co. immediately following his graduation from Lehigh University in 1934 with a B. S. degree in Mechanical Engineering. He started in the experimental laboratory of the company, and early in 1942 was appointed manager of the testing laboratory.

The IMPORTANCE o
in COLD STARTING o

BEFORE any attempt is made to discuss the various types of accessory equipment used as aids to cold starting, it will be assumed that under all conditions noted the fuels, lubricants, batteries, and cranking equipment have been selected to provide what may be considered normal operation and cranking ability at the temperatures stated. This will be necessary to eliminate any complexity of variables that naturally will affect the starting ability, and to reduce the problem concerning accessory equipment to those appliances not normally used for starting above freezing temperatures.

Naturally, under subnormal temperatures, the resultant heat of compression under the initial cranking conditions does not provide a temperature sufficiently high for self-ignition of the fuel (at least 575 F for a 43 cetane fuel). Therefore, the prime purpose of most of the accessory equipment is artificially to heat the incoming charge of air sufficiently to obtain combustion. This naturally reduces the amount of energy required from the cranking system. Basically, there are two methods of heating this incoming charge of air: either preheaters (engine motionless) or cranking heaters (heat applied during cranking) –

1. Electrical

(a) Resistance coils located in the inlet manifold – current supplied by starting batteries. May be preheaters or combinations of preheaters and cranking heaters, but never cranking heaters alone.

(b) Immersion heaters located in the cylinder-block coolant – current supplied by external source. Classed as preheaters.

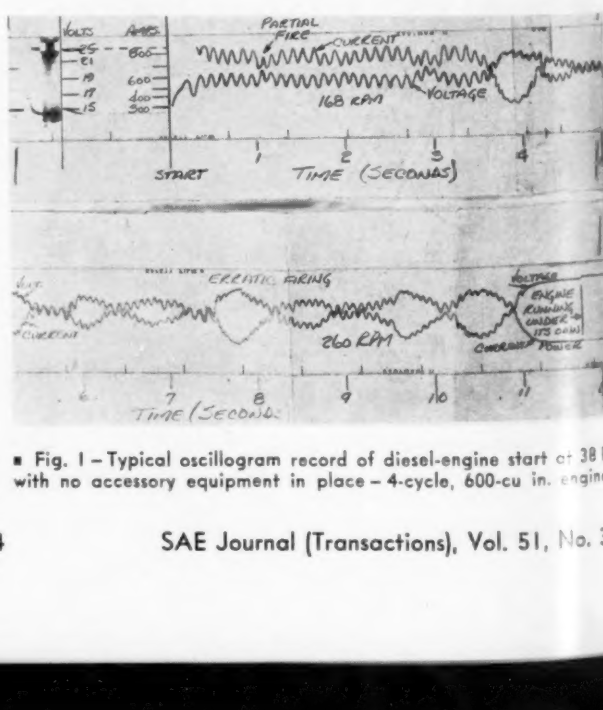
2. Combustion of fuel (diesel)

(a) In the inlet manifold. This may be either a preheater or a cranking heater.

(b) In a separate compartment (pot-type stove) interconnected with the cylinder-block coolant. This is a preheater and bears a close relationship to the electrical immersion heater as covered above.

In studying the effects of any auxiliary equipment in promoting cold starts, it may be well first to determine the engine starting characteristics without such aids at its critical temperature. The conditions encountered may be noted by the variations of the voltage and current supplied to the starting motor during the cranking period. Variations in these quantities reveal the changes occurring in the engine. By utilizing suitable recording instruments, such as oscillographs and movie cameras, the exact values of the electrical changes, as well as the trends in cranking speeds, can be obtained. Fig. 1 shows the starting motor voltage and current record of a typical start of a 600-cu in. engine at 38 F, as recorded on a moving film. It will be noted that the breakaway current was in excess of 800 amp, because the trace appears on the film after approximately one-third second. The wave form of the trace of both voltage and current shows the fluctuation in torque with each compression; therefore, three complete cycles of this wave represent one revolution (6-cyl, 4-cycle engine). The first noticeable deviation in both the current and voltage occurs after approximately 1 sec of cranking, and indicates a partial combustion. A similar condition occurs after approximately 3 sec of cranking, and a definite surge of the engine is noted at the 4-sec point. This surge appears rather consistently during the remaining cranking period, indicating the continuation of erratic combustion, until after 11 sec the engine fires regularly and is able to run under its own power.

The application of ribbon-type nichrome resistance coils in insulated blocks, mounted as close to the combustion



■ Fig. 1 – Typical oscillogram record of diesel-engine start at 38 F with no accessory equipment in place – 4-cycle, 600-cu in. engine

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

# of ACCESSORY EQUIPMENT of DIESEL ENGINES

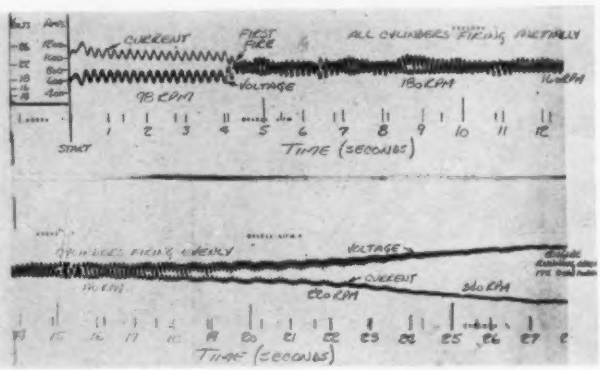
by W. J. PELIZZONI
International-Plainfield Motor Co.

chamber as possible to prevent excessive air heat loss, is shown in Fig. 2. The ends of the nichrome ribbon are fastened to copper binding posts which are used with copper jumpers to connect the series of heating elements. The elements are energized by suitable solenoid switches controlled by a pushbutton located conveniently for the operator.

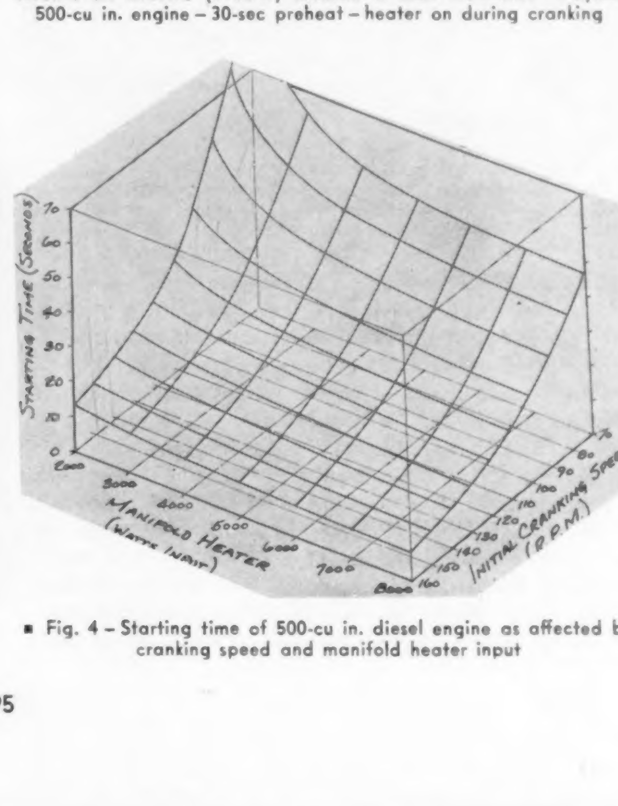
Usually, the elements are preheated before any attempt is made at starting to allow sufficient saturation of the resistance coils and a build-up of heat in the air adjacent to the coils. Preheat time, of course, will have some effect on the resultant start, but it should be remembered that the drain on the batteries may offset any possible gain which otherwise may have been obtained, by a resultant lower cranking speed. The oscillogram shown in Fig. 3 represents a start obtained on a 500-cu in. engine using a 30-sec preheat period, wherein the heaters remained on during the cranking time. The heat input of 3730 w during the preheat period is reduced by about 25% during the cranking period because of the additional drain on the battery and lower available voltage. The first fire occurs after approximately 4 sec, and partial firing in every cylinder continues very evenly thereafter until the engine is able to run on its own power. During the initial partial firing there will be noted some slight change of engine speed, but after approximately 15 sec the speed seems to accelerate rather steadily.

Since cranking speed plays a definite part in providing increased compression pressures and temperatures, particularly at the lower speeds, a relationship between cranking speed and manifold heater input (also affecting maximum compression temperatures), as related to total starting time appears in Fig. 4. The benefits of the electrical heaters at capacities above 4000 w appear rather insignificant as compared to an increase in cranking speed. For example, at 100-rpm cranking speed, a change from 4000 to 8000 w in heater input will reduce the starting time approximately 5 sec, whereas, under the same initial conditions, an increase in cranking speed to 120 rpm will reduce the starting time by approximately 10 sec, or 30%.

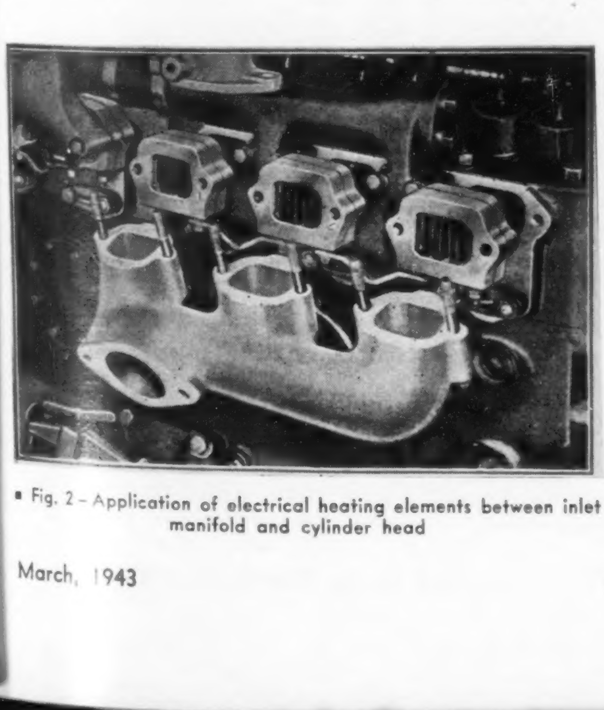
Starting-battery energized electrical heating elements



■ Fig. 3 – Oscillogram record of diesel-engine start at – 1 F, using electric air heaters (3730 w) located in inlet manifolds – 4-cycle, 500-cu in. engine – 30-sec preheat – heater on during cranking



■ Fig. 4 – Starting time of 500-cu in. diesel engine as affected by cranking speed and manifold heater input

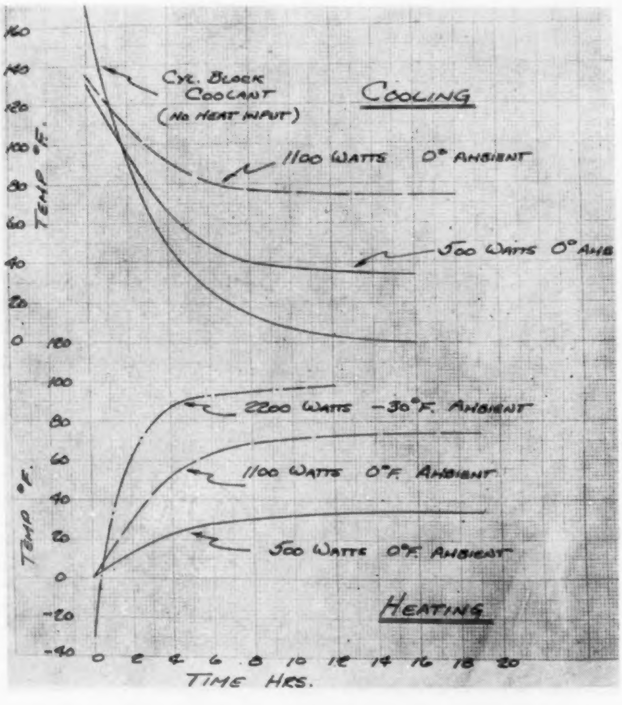


■ Fig. 2 – Application of electrical heating elements between inlet manifold and cylinder head

naturally penalize the available capacity remaining for cranking, particularly when heating during the cranking period. In some cases it has been found beneficial to preheat only, allowing full battery capacity for cranking. This results in increased initial speeds which, as shown above, probably will reduce the starting time. Aside from the above, electric heating of this type is satisfactory because of its simplicity and low maintenance requirements.

Cylinder-block coolant immersion heaters of the electrical resistance type are also ideal from the standpoint of simplicity and maintenance requirements. In this case, however, the units can only be used with the usual house current (110 v), because of the high output and length of heating time required for beneficial results. Furthermore, their use is restricted to any location where suitable power source is available.

This subject is introduced here because of its similarity to another type of heater which will be discussed later. The heating characteristics of the engine for both are practically identical. Fig. 5 shows a comparison of cylinder-



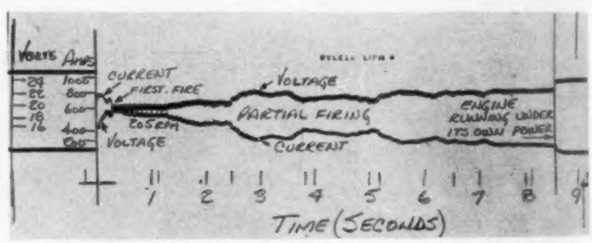
■ Fig. 5 – Cooling and heating rates of engines with 6-gal coolant capacity – electrical immersion heaters of capacity as indicated

block coolant temperatures and ambient air temperatures. The resultant cylinder-block-coolant equilibrium temperature under any specific heat input rate will naturally depend on the surface area of the engine, the capacity of the cooling system, the total engine weight, and the air circulation rate. It will be noted that in an engine of 6-gal coolant capacity, approximately 12 hr are required for reducing the temperature from 170 F to the ambient air temperature, which in this case was 0 F, and which had an average velocity of 6 mph. Immediately after the engine stops operating, the use of immersion heaters of this type will retard the cooling-down rate, and equilibrium temperatures will be reached in a shorter elapsed time. On the heating cycles using immersion heaters, on the other hand, a nearly equilibrium temperature is reached in approximately the same time, again depending on the heat input. As shown, under 0 F ambient air, coolant temperatures of 30 F and 70 F are attained after approximately eight hours of operation by the use of 500-w and 1100-w heaters, respectively, whereas under an ambient air temperature of −30 F, with 2200 w a block temperature of 90 F will be obtained after only 4 hr of operation. Regardless of the ambient air temperature, when a cylinder-block coolant temperature of at least 60 F exists, a start can be obtained in a reasonable time without the use of any additional heaters.

This can be accomplished because of the relatively high air temperature existing in the cylinder and the resultant higher cranking speed allowed by the reduced piston-to-cylinder-sidewall friction.

By comparison of the heating and cooling cycles shown, it will be noted that the equilibrium temperatures under similar heat inputs will be equal after sufficient elapsed time. The advantage, however, is in the cooling phase because the cylinder-block temperature is favorable before the equilibrium temperature is reached. Therefore, it is important that heaters of this type be applied as soon as the engine stops operating so that good starting conditions prevail at all times.

Fig. 6 shows the oscillogram record of current and voltage of a start accomplished under 0 F ambient air conditions and 74 F equilibrium temperature of the cylinder-block coolant as obtained through the use of an 1100-w immersion heater. The interesting points brought out in this graph are the high initial cranking speed and



■ Fig. 6 – Oscillogram record of 500-cu in. diesel-engine start at 0 F – immersion heaters (1100-w total capacity) in place and cylinder-block coolant at equilibrium temperature of 74 F

early first-fire of the engine. The conditions under which this start was made were comparable to those as shown in Fig. 3 except as stated (no manifold heaters). It will be noted from this comparison that the breakaway torque and initial cranking torque have been reduced at least 25% by the increased block temperature, which is reflected as an increase in initial cranking speed of at least 25%. As previously shown in Fig. 4, this change alone will reduce starting time considerably, but with the increased air temperature and quick first-fire, the starting time is reduced about 70%.

Electric heaters, as covered above, may be considered satisfactory providing the conditions existing warrant their use. In cases where the installation and type of engine require an extremely rapid and high-capacity air heating arrangement, it becomes necessary to depart from the electrical method and adopt some other device which will produce the necessary air temperatures without overtaxing the cranking equipment. One suitable and fairly popular

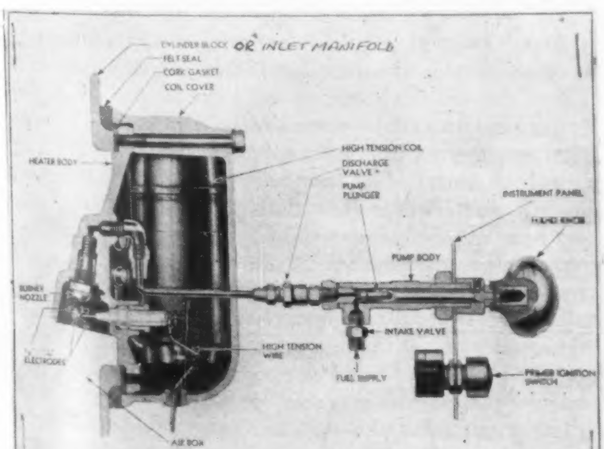
method of obtaining this high Btu output is to burn fuel oil directly in either the intake manifold or air box of the engine. Provided the fuel burner is suitably selected and properly adjusted, the benefits of an output as high as 2000 Btu per min can be obtained; and if there is sufficient excess air in the induction system during the flame-burning and cranking period, a relatively easy start will be obtained.

Generally these flame-throwing air heaters consist of a small pressure oil burner with electric ignition similar in principle to the burners now common in domestic furnaces. The burner proper can be mounted either in the inlet manifold or in the engine air box and obtain its air either from the natural flow of air through the manifold during cranking or from the charging blower for unsupercharged and supercharged engines, respectively. In both cases the products of combustion along with the heated excess air will be circulated through the engine cylinders and the necessary resultant heat for normal combustion obtained.

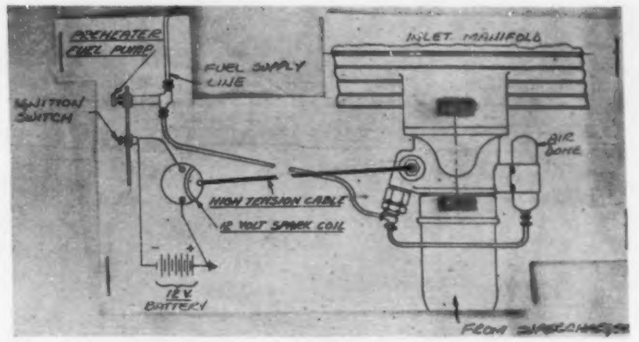
The greater success of flame throwers as starting aids on supercharged engines, in addition to the greater excess air capacity, might possibly be due to the better chance of continuity of flow created by the increased valve overlap and extended inlet timing.

As shown in Figs. 7 and 8, the complete unit consists mainly of two basic parts. The pressure pump and ignition switch are usually mounted as an assembly on the dash or where they are convenient to the other engine controls. The vibrator ignition coil and points and nozzle burner comprise the other assembly which becomes a part of the engine by its mounting on the inlet manifold or air box. A small air-dome reservoir (not shown) is provided between the priming pump and the nozzle to prevent dribbling and to provide a more nearly constant pressure against the nozzle. Depending on the calibration of the burner unit and the type of air supply, it may be necessary to incorporate both a fuel and air control valve to allow adjustment for a proper mixture. This type of unit is a cranking heater because the air supply necessary to support the combustion of the fuel can only be obtained through the natural flow created during engine motion. Likewise, the proper distribution of the heat of this combustion can only be accomplished through this engine motion.

Depending on the temperature and cranking speed of



▪ Fig. 7 – Flame primer as used for preheating the ingoing charge of air to the cylinders



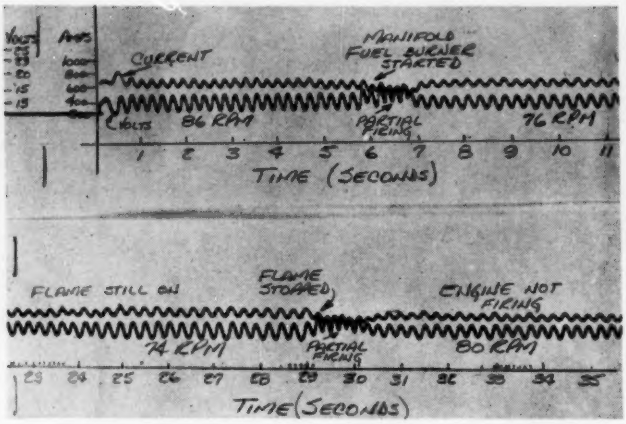
▪ Fig. 8 – Electrical system of flame primer

the engine, a start may be accomplished immediately with the above arrangement, providing there is sufficient excess air remaining in the cylinders to support combustion of the fuel normally injected. In the cases where either sufficient excess air is not present or the cranking speed is too low, it will be necessary to continue cranking while the flame thrower is ignited to obtain the required beneficial results. Thus, the cranking motor merely assists the weak combustion, until sufficient speed is reached, at which point the engine is able to operate under its own power and the auxiliary fuel-burning equipment may be shut off.

Generally, the application of the inlet manifold flame-thrower type of heaters is not successful on unsupercharged engines because of the lack of sufficient excess air. Undoubtedly, the fuel-burning rate could be calibrated so as to work satisfactorily at one particular cranking speed; that is, the fuel-burning rate could be selected so as to utilize only a specific part of the available air, but this becomes impractical since the amount of excess air available in unsupercharged engines is relatively small and any slight speed change would require a fine adjustment of the fuel control. A typical attempted start under the above conditions is shown in Fig. 9. It will be noted that during the initial cranking speed of 86 rpm at the −2 F temperature, the conditions were fairly constant, but at the instant the flame thrower started the combustion was apparently carried on into the cylinders where partial firing occurred. Continued use of the manifold flame, however, eliminated any further partial firing, evidently because of the lack of sufficient oxygen remaining to support combustion in the cylinders. This condition remained practically the same for the entire 23 sec that the burner was on, but as soon as the burner was turned off the cylinders showed a partial firing, evidently because the proper mixture was again momentarily obtained.

This illustrates how sensitive a control would be required to obtain reasonable starts. Another interesting point to note is the slight reduction in cranking speed during the flame-burning period, which probably is caused by the reaction of compression against the burning charge.

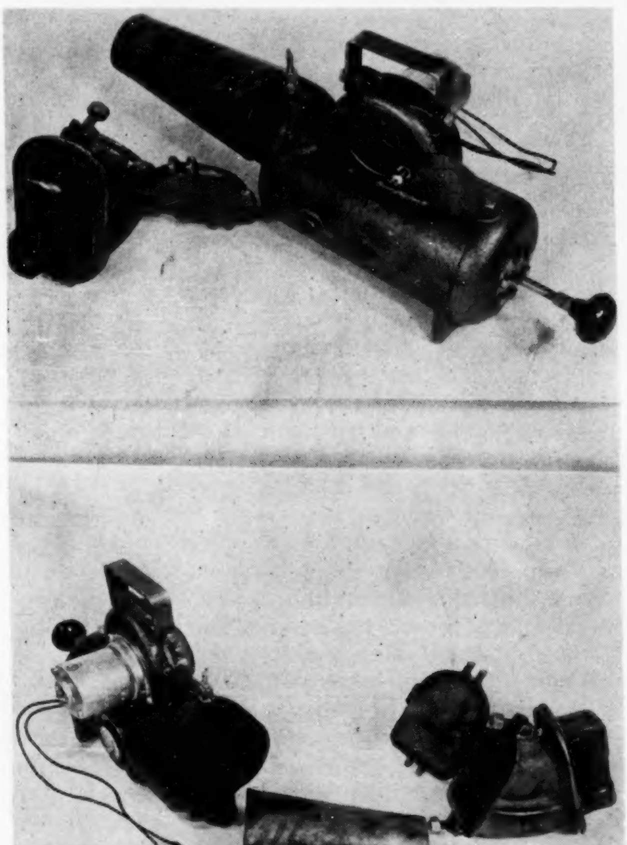
Another type of fuel-burning arrangement, which is used only as a preheater and is similar to a blowtorch with an air circulating fan, is shown in Fig. 10. The photograph indicates the manner in which the unit containing the pressure tank, fuel supply, and circulating fan is adapted to the engine air box or cylinder block. It will be seen that the adapter with a trap-door cover is designed so as to provide a directional flow of hot air with an escapement port directly underneath. This unit requires an outside source for ignition and may be ignited and thoroughly



■ Fig. 9 – Attempted start of unsupercharged engine using fuel burner directly in the inlet manifold, — 2 F engine and ambient air temperature – 500 cu in. engine

warmed before it is put in contact with the engine. The fuel-adjustment needle gives the desired burning rate required to match the air flow. Under a —40 F ambient air and saturated engine temperature condition, the cylinder-head coolant temperatures will increase even faster than that shown on Fig. 5, wherein the 2200-w electrical immersion heater produced a temperature of 60 F from —30 F in 2 hr. With this unit, a temperature of 60 F from —40 F may be obtained in less than 1 hr.

It might be possible to adapt this type of heater to any engine, and circulate the hot air through the crankcase.



■ Fig. 10 – Fuel burner and blower arrangement for preheating the engine cylinders

The only major objection to such usage would be the possible resultant condensation which would form on the inner surfaces of the engines; otherwise it would prove just as effective as the air box application.

A separate gasoline or fuel-burning stove which has a heat exchanger interconnected with the cylinder-block coolant may also be utilized very effectively. This type of heater consists mainly of an adjustable metering device which feeds the fuel into a vaporizing chamber. Ignition of the fuel in the vaporizing chamber is accomplished by some outside source such as a lighted match or taper. The heat of combustion is utilized by circulation of the flame in the fire box around the boiler. Thus, through the temperature differential existing between the coolant in the heater and in the cylinder block, thermosyphon flow provides the necessary circulation. The design of one small unit of this type is such that the fuel burning rate can be varied from approximately 1200 to 6000 Btu per hr.

The cylinder-block temperature rise with the separate fuel burning heater will be similar to that as noted by use of the electrical immersion heater. (Refer to Fig. 5.) For an equal Btu input, however, the resultant temperature rise will be less because of the lower efficiency of the burner and the circulation necessary through pipe connections. Again, the importance of time with the use of either the immersion-type heater or thermosyphon heat exchanger unit is stressed, together with the advantages obtained by application of this heat as soon as the engine stops operating.

In addition to the various methods of assisting subnormal temperature starts on diesel engines as explained above, there are various applications of the same type units which achieve a similar net result by some slight modification of the principles. Still other engine manufacturers include mechanical changeover mechanisms, which either increase compression materially or decrease compression and utilize gasoline (by carburetion) spark ignited in the cylinders. No mention was made of glow plugs, which are an electrical resistance type of unit placed either directly in the combustion chamber or in some antechamber to provide an actual hot-spot ignition of the injected fuel. Generally speaking, this type of unit will not withstand the temperatures attendant in the combustion chamber of high-speed engines and have been found a continued source of trouble by burning out.

The various methods of applying additional heat described herein, as required for aids in cold starting, all seem to provide the necessary boost in temperatures to relieve the cranking system of a large portion of the demand which would otherwise be required. The electrical systems, as well as the fuel-burning systems, have their limitations which naturally are affected by either ambient air temperatures, cranking ability at these temperatures, simplicity, adaptability, time necessary for effectiveness and maintenance. It is believed that no single arrangement would be sufficient to meet all the above requirements at subzero temperatures; therefore it might be necessary to incorporate either a combination of these units or a combination of one accessory with some special fuel additive.

The writer takes this opportunity to express thanks to the various engine manufacturers for their cooperation in submitting data. This information was freely given in spite of the demands placed upon them by their intense occupation toward the war effort, and is greatly appreciated.

CONVERSION of AUTOMOTIVE PARTS PLANTS to War Production

by JOSEPH GESCHELIN

Chilton Co.

CONVERSION of the entire automotive industry to the production of the weapons of war is a phenomenon which still is not completely understood by many outside its ranks. For one thing, even those who normally deal with the industry do not fully appreciate that the automotive industry is an integration of a relatively small number of organizations which serve as the prime vehicle producers; and that these companies – final assembly lines, if you will – are served by a huge aggregation of parts and accessories producers, body builders, and suppliers of every kind, representing an imposing total of capital investment and employment.

Conversion of the major vehicle producers has been a topic for discussion during the past two years. The author has presented the picture at various times at section meetings of the Society; and more recently covered the latest aspects of the situation in an article entitled "Conversion Almost Complete" in the Oct. 1, 1942, issue of *Automotive and Aviation Industries.*

It is significant, however, that all public discussion has centered about the vehicle producers, except in so far as the picture has included divisions of the larger corporations, and no distinctive place has been given to the activity of the parts makers. It is our purpose to record the accomplishment of this branch of the automotive industry, to complete the story more adequately.

To supplement the author's first-hand knowledge of industry operations, we circularized a cross-section of the parts and accessories manufacturers. The cooperation of the group was splendid; their statistics and comments have made the following presentation possible.

In the first place, it may be safely said that, as a group, the automotive parts makers are as close to being 100% on war production as any industry group can be. By the very nature of their business, there are some whose contribution to the war effort must be made by catering to the service needs of the public, of private transportation, and so forth, so that the actual percentage of their effort directly for the military services may range from 100% to as low as 80% in a few instances. By far the majority is contributing from 95 to 100% of activity to the war effort.

When the parts industry is examined more closely, it is obvious that in the main its members were among the earliest to concentrate on war production; and that they attained their peak sooner than did the primary vehicle producers. On the basis of actual reports, a fair proportion of parts makers have been doing work for the Armed Services for three years; the majority have been engaged for at least 18 months; and only a few have come into the picture within the past six months.

The parts industry is so diversified as to defy generalization, except in one respect. It may be said safely that the parts industry as a whole constitutes the best example of well organized "subcontracting" in our economic system. Certainly it was no new experience for this group to participate in the national call for subcontracting, albeit the experience was new for many companies outside the automotive fold.

Let us examine the major elements of the parts industry so as to establish a sound basis for further analysis. Here are a few of the categories in which the parts makers may be classified:

1. Producers of heavy-duty axles, clutches, and transmissions, for motor trucks, buses, tractors, and so forth
2. Producers of axles, clutches, and transmissions for passenger car use
3. Commercial engine builders for truck, tractor, bus, marine, and industrial use
4. Body builders
5. Spring makers for engine springs, chassis springs, accessories, and so forth
6. Producers of pistons, piston rings, cylinder liners
7. Manufacturers of engine valves and valve lifters for passenger cars, trucks, tractors, marine engines, airplane engines, and so forth

---

THE AUTHOR: JOSEPH GESCHELIN (M '21) served as an engineering editor of *Automotive Industries* from 1930 to 1936, and since then has been technical editor in Detroit for all the Chilton Publications. A graduate of Cooper Union Institute of Technology with a B.S. in Mechanical Engineering, he gained experience in jobs in motor truck and parts plants that combined practical engineering and sales promotion, which ultimately prepared the way for technical editorship. Mr. Geschelin, who has been consultant to the Bureau of Ships, U. S. Navy, in addition to his editorial work, has been a member of the Production Activity since 1930 and was vice-president representing this Activity in 1942.

---

8. Manufacturers of electrical equipment – starting motors, generators, relays, horns, instruments, spark plugs, wiring harness, batteries, ignition devices

9. Producers of the variety of accessories used on passenger cars, trucks and buses – lamps, signaling devices, windshield wipers, visors, cigar lighters, mirrors, and so forth

10. Manufacturers of brake systems

11. Carburetor specialists

12. Drop forging, stamping, die casting, plastic molding, and a host of other specialties

13. Specialists making radiators and heaters

14. Producers of trim items, hardware, moldings, upholstery, and so forth

15. Bearing manufacturers

16. Wheels and brake drums

17. Radios

Doubtless the list could be extended still more. But the foregoing at least gives some impression of the variety of activity upon which the vehicle producers have drawn.

When you scan this list, it becomes obvious why the parts industry was able to make such remarkable showing in the war effort. For one thing, in the great majority of individual operations it was evident that the war effort would best be served by continuing with the production of the same character of product. To cite specific examples, it may be noted that a producer of engines, engine valves, crankshafts, pistons, piston rings, electrical equipment, carburetors and so forth could serve best by continuing the same specialty.

True, an adjustment in viewpoint had to be made. With the stoppage of passenger car production, the energy of a given organization had to be turned to serving military airplane requirements, engines for the Navy, parts for tanks, tractors, amphibian vehicles. Production was shifted to larger versions of the usual product, to heavy-duty models. Major complication in this transitory stage was the fact that in total units of output the requirements were low as contrasted with the needs of an industry building up to 5,000,000 motor cars a year. In many instances this caused a shift from mass production, in the terms known within the automotive industry, to job-lot production. This shift has increased unit cost materially although the dollar volume is many times greater than normal.

■ Conversion to Military Needs

By the same token, producers of other types of peacetime accessories for the industry found themselves without a market save for a limited volume of service requirements. This group set about to convert radically for the production of war products completely outside their normal experience. Even here, a proper appraisal of the situation made it evident that the conversion should proceed along the lines of the "know-how" of the organization and within the limits of existing equipment and facilities.

Thus we find the major body builders engaged in the making of elements of big airplanes – wings, tail surfaces, engine nacelles, fuselage sections, for example. Others have undertaken the making of guns and gun mounts, fuses, cartridge cases, and the like. Some are making bombsights, gyroscopes and gyrocompasses, machine tools, propellers.

The accelerated tank program, the airplane program that has given us mastery of the air, the Navy program: these combined to make imperative an extensive increase in the facilities of most of the parts makers. Airplane rings are required by the millions, electrical equipment, spark plugs, instruments, valves, and a host of other critical items in an internal-combustion war strained the capacity of producers, made it necessary to create additional plants and new facilities.

We might pause at this point to recall the vain and foolish attempts of the uninitiated to force conversion of this industry along lines completely inconsistent with its knowledge and facilities. The record as it is unfolded will prove unmistakably the soundness of the program finally developed by automotive producers.

So much for the overall perspective. A more intimate background may be gained by considering the cooperative role of the industry as expressed by the work of the Automotive Council for War Production, the Society of Automotive Engineers, the parts associations such as the Automotive Parts and Equipment Manufacturers Association, National Standard Parts Association, Motor and Equipment Manufacturers Association, and the National Automotive Parts Association. These organizations, each representing a specific cross-section of the industry's activity, have made a remarkable contribution to the war effort, far beyond the appreciation of any except those who were directly concerned.

But in addition to the work of such organizations there has been an integration both of individual efforts and of the contributions of smaller and more specialized groups. To these may be added the top ranking representatives of many individual companies who were released to serve with the War Production Board, with other Governmental agencies, and with the Armed Services, either in civilian capacity or in uniform.

Out of the many engrossing examples of conversion and industry cooperation let us consider just a few chosen at random from the record contributed for this paper. Please remember that this is but a tiny sampling of the whole. Neither time nor space would permit us to do justice to even a small percentage of the organizations who are now all-out for the duration.

Consider first some typical axle and transmission builders. The Timken-Detroit Axle Co. is proud of a record of cooperation with the U. S. Army on problems of motorization and mechanization dating back to World War I. This has been effected through a company policy of cooperation with U. S. Army services, through advisory committees such as SAE-Quartermaster and SAE-Ordnance groups.

Directly as a result of this intimate contact with the U. S. Army, Timken was tooled up and had available for sale to prime contractors front driving axles, rear driving axles, tandem rear-axle units, transfer cases, gun carriage axles, and final drive units for tanks.

At Timken the conversion process was simply that of switching from one type of product to another by more intensive use of existing facilities, by the introduction of some new equipment, but mainly by the establishment of a widespread subcontracting set-up, which made it possible to satisfy a tremendous increase in demand for wartime products.

Timken has contributed to the development of drive elements for military vehicles and combat tanks, not only in the design phases but in the construction of pilot models for the Army's proving grounds. Effective Jan. 1, 1942, Timken signed a license agreement with our Government

granting the use of certain patents and improvements thereon, on a royalty-free basis terminating with the conclusion of the emergency.

Spicer Mfg. Corp. supplies a similar experience. For many years this organization has worked with Holabird Ordnance Motor Base on motor transport vehicles, and with Army Ordnance in Washington and at Rock Island Arsenal on combat vehicles. Spicer's war activities started about two years ago, in scope embracing the production of propeller shafts and universal joints, front and rear axles, transmissions, transfer cases, clutches, power take-offs, and so on.

These war products are of precisely the same general character as Spicer's normal production, with difference in size, quantity, and detail. This made it possible to utilize all of the existing equipment, the change-over being effected by a reorganization of plant layout, by the acquisition of many items of new equipment. In addition, Spicer has completed a major plant expansion program designed to provide facilities for the mass production of light-tank transmissions and torque converters.

In the case of Timken and Spicer, the immediate conversion to war production was a direct result of specialized knowledge of engineering, manufacturing, and management.

Axle Producers Major Contributors

In similar fashion, we find that the other major producers in the field of heavy-duty drive elements such as the Axle Division, Eaton Mfg. Co. and Clark Equipment Co. have made a major contribution to the Army's motorization program. In the case of both companies, there have been major expansion programs in the interest of accelerating the output of essential units. Clark Equipment, for example, has built an entirely new and specialized plant for the production of heavy-duty axles and axle housings.

Another example in a related field is that of the Twin Disc Clutch Co., producer of a variety of power transmission devices for machine tools, for agricultural tractors, and for all forms of machinery using internal-combustion engines. Conversion to war production in this instance was simply a matter of expansion and subcontracting to handle the demand for a greatly increased output of essentially the same products.

A short time ago Twin Disc developed a line of hydraulic transmission units which now will be built exclusively for military vehicles. To handle this program the company has undertaken a major expansion program in its Rockford plant.

Consider now a brief comment on what has happened to manufacturers of automotive wheels and bodies. Kelsey-Hayes Wheel Co., one of the best known names in the industry, is devoting practically 100% of its facilities to war production. Among its current products are machine guns, projectiles, cylinder barrels for radial airplane engines, oxygen cylinders, wheel assemblies and other parts for tanks, airplane parts, armor plate castings, axle housing castings, wheels, hubs, brakes, gun carriage equipment, and so forth.

In the main, the character of parts for war production is deliberately such as to make possible the utilization of facilities and equipment formerly used for automotive production. However, the greatly increased volume has made it necessary to enlarge plant facilities materially. Among other things, the Kelsey-Hayes foundry was converted to the production of steel castings and armor plate castings.

Motor Wheel Corp., another of the leading names in the parts field, is now 100% on war work, including projectiles, wheels, hubs, centrifuse brake drums, and other stamped parts, bogie wheels and idler wheels for tanks, and so forth. In addition, the company is doing a great amount of subcontract work for builders of machine guns and cannon. One of its major contributions has been the development of centrifuse brake drums for airplanes.

Motor Wheel's conversion problem has been a typical one, since they found it possible to use existing facilities in the main, with only slight plant additions and a moderate addition to the production equipment. The wheel assembly lines and huge press department have been adapted almost entirely to war production. Again this illustrates the general principle of a conscious selection of those items of war production best suited to the existing facilities and within the knowledge of the organization.

What is the picture in a typical automotive body plant? The Murray Corp. of America furnishes an excellent example. In this instance we find a radical shift from body making to an entirely new line of products, employing similar equipment and similar techniques. Among the items currently produced for the war program are the following: airplane wings, airplane wing tips, airplane nacelles, stainless steel assemblies, frames for jeeps, combat cars and military vehicles, parts for AA searchlights, "muzzle brakes" for machine guns.

According to Murray, although airplane work and automotive body building are quite dissimilar there are many techniques common to both. In the transition to airplane work, Murray has pioneered many adaptations of methods which have come into common use, largely due to a transfer of experience in steel fabrication to the problems of the forming and assembly of aluminum parts. Initially, Murray employed the same methods and tooling used by the aircraft industry. But with experience automotive methods and equipment began to be adapted freely. For example, the use of draw dies in heavy presses instead of power hammers increased output of some parts as much as fifteen-fold.

Application of draw dies to the formation of aluminum sheets was made possible by the development of the "ice-box" method of delayed aging of heat-treated sheets, stemming from the earlier method of storing aluminum alloy rivets developed by the Aluminum Co. of America.

One of the chief contributions of the automotive industry to aircraft production has been the development of precision jigs, each built to a master jig and all identical, resulting in a great degree of interchangeability so vital today when plants in different parts of the country contribute parts for assembly at widely separated airplane plants.

The frame division continues its automotive specialty without much change since the product is essentially of the same character.

Briggs Mfg. Co. is 99% on war work, producing such items as a wide variety of airplane wings; aileron, wing tip, wing flap, stabilizer fin, rudder and tail assemblies; airplane bulkheads; door assemblies including bomb bay doors; bomber duct assemblies; bomber turrets; shells and cartridge cases; tank hulls; tank turrets; airplane-engine parts; non-ferrous castings.

Actually, Briggs is producing weapons of war the general character of which differs radically from the peacetime activity of building automotive bodies and plumbing. Nevertheless, Briggs had the knowledge as a heavy-duty sheet metal specialist; had the heavy-duty press equipment and other mass-production techniques which could be readily adapted to the new problems. In addition, the company had been a leader in the development of welding techniques and welding equipment. And this experience was found invaluable in its application to the manufacture of airplane assemblies and tank hulls.

Now we have in the industry an important group of specialists making piston rings. These include such names as Muskegon Piston Ring Co., Ramsey Accessories Mfg. Corp., Perfect Circle Co., McQuay-Norris Mfg. Co., American Hammered Piston Ring Division, Koppers Co., Sealed Power Corp., Hastings Mfg. Co., and many others. To these organizations the war program simply meant an intensification of activity at the same stand. However, the war in the air produced a demand for millions of airplane-engine rings which in size and specifications created an entirely new problem for the major automotive producers.

In two instances – Muskegon Piston Ring and Ramsey – the addition of airplane rings created a need for new facilities. In the case of the former, it resulted in an expansion of the existing plant; in the latter, Ramsey outfitted a new plant in Muskegon, devoted exclusively to the manufacture of airplane rings. The Perfect Circle Co. expanded its facilities and intensified its output at first; then embarked on the construction of an entirely new plant in Richmond, Ind., which was recently completed.

Sealed Power is engaged in the manufacture of many allied products, including pistons, cylinder liners, wristpins, and other accessories for internal-combustion engines. Its piston-ring facilities have been greatly expanded to take care of the war program with its heavy quota of airplane rings.

Muskegon Piston Ring also has expanded the operations of its foundry division in Sparta. This foundry, as is well known, has served as a supplier of gray iron castings for automotive and airplane rings to many other piston-ring manufacturers. A year ago this foundry had no facilities for producing bronze piston-ring castings, which are used for oil seals in aircraft engines and in propeller mechanisms. Since then they have installed a battery of bronze-melting furnaces, supplying not only Muskegon but several other ring manufacturers.

There is an interesting human interest story concerned with the piston-ring producing group. It is well known that up to a relatively short time ago the art of making rings was a closely guarded secret. Actually there was but little exchange of information so far as manufacturing methods were concerned. But with the incidence of the war the picture changed completely.

The heavy and unprecedented demand for airplane rings taxed the capacity of the specialists, made it necessary to draw other producers into the picture. The industry responded patriotically to the call of the Armed Services. The leading producers banded together in the common cause, pooled their hard-won technical information on design and manufacture, made it available to the others. In the process, representatives of the group visited other plants and were given blueprints, production information, and even exchanged manufacturing equipment. Troubles were discussed freely within the group and one helped the other to overcome them.

In the case of the Army Transportation Corps, a group of independent ring manufacturers were appointed as a committee to make recommendations to the Army on a standardization program leading to the development of a set-up for the types of rings and kind of oversizes which would be to the best advantage of the Army's maintenance program.

Here indeed is an exemplary instance of unselfish cooperation in a common cause.

Obviously neither time nor space will permit us to continue the unbroken chain of specific cases so as to do full justice to the participation of the parts industry in the war effort. Certainly it would be impossible to outline the activity of the great Borg-Warner Corp., and of its subsidiaries such as Warner Gear Division, Mechanics Universal Joint Division, Borg & Beck, and others. Neither can we do more than touch briefly on the many independent parts makers and the other corporation groups such as the Eaton Mfg. Co., Clark Equipment Co., Fuller Mfg. Co., Electric Auto-Lite Co. and its subsidiaries.

■ Conversion Record of GM

General Motors Corp., speaking for all of its divisions, advises that the corporation as a whole is about 95% on war work. In some divisions the record is even better. For example, Rochester Products, Delco Radio, New Departure, and Harrison Radiator are 100% on war work.

The majority of the parts divisions of General Motors have been able to convert to war production without major plant expansions, although in some instances a small wing or addition has been built. Where major expansion has taken place it has been due to the requirements of products completely foreign to the former activity. One example of this is the new aluminum foundry at Delco-Remy; another is the machine gun plant at Saginaw Steering Gear and at AC Spark Plug. At Rochester Products, airplane electrical equipment has completely replaced items formerly built for automobiles. Generally speaking, however, it may be said that each of the divisions is producing at least certain items which are closely akin to peacetime products.

Saginaw Malleable Iron Division recently converted to steel castings and is specializing in the production of ArmaSteel, an alloy which has found its way into many products made by the other divisions.

AC Spark Plug has employed ArmaSteel in machine guns, reducing the number of types of steel in the gun from 44 to 15, eliminating many drop forgings in the process. Here again is an example of cooperative effort in a common cause. According to George Mann, Jr., general manager, AC Spark Plug Division, all of the better methods, improvements, and new ideas have been freely passed along to other gun producers and to the Government. Just recently, representatives of two new machine gun facilities came to AC for assistance. They were supplied with drawings for all tools, jigs, fixtures, and gages required in the program, and were given other help in the interest of getting the new projects going without delay.

New Departure is 100% on war work. While still engaged in the manufacture of ball bearings, this division has had to accommodate many new types and sizes which were completely out of the range of automotive experience.

Although the existing equipment was found adaptable, it was necessary to retool more than 50% of it. This division is currently producing about 18,000 different varieties of bearings.

Harrison Radiator Division has had to undergo a major shift in its emphasis upon certain of its specialties. For example, Harrison was making oil coolers, cellular radiators, water coolers, air coolers, Prestone radiators, supercharger intercoolers, heavy-duty radiators, and so forth, before Pearl Harbor. However, the items that constituted mass production – cellular radiators, thermostats, heaters, and defrosters – now account for only 2% of production. On the other hand, the tubular radiator, which has represented but small volume since 1938, has been largely expanded to handle heavy-duty radiators for trucks, tanks, and armored cars.

Conversion has made it possible to utilize all the buildings and much of the equipment used in normal automotive production. To this have been added some floor space, considerable equipment, and extra facilities to take care of military requirements.

Bendix Aviation Corp. is devoting about 97% of its activity directly to war production, specializing in items such as the following – carburetors, brakes, airplane landing gear, hydraulic units, Bendix-Weiss constant-velocity universal joints, ordnance materiel, airplane gun turrets, and so forth.

Many of the items on the list were made in peacetime production and constitute a major expansion of normal activity, augmented by new facilities now in operation. On the other hand the purely automotive products of the corporation constitute a major contribution to the war program, since their development and manufacture have been increased to meet the demands of the military services.

Monroe Auto Equipment Co., well known as an automotive parts producer, particularly in such specialties as direct-acting shock absorbers and sway bars, has turned its energies to the making of projectiles, combat tank seats, and so forth. In fact, shock absorbers are the only peacetime product still in the picture. Monroe has made several major additions to its facilities to take care of the tank seat and projectile program.

W. C. Lipe, Inc. is 100% on war work, producing heavy-duty clutches for tanks, an exclusive development; heavy-duty clutches for military vehicles, and many items of machine tool equipment.

Firestone Steel Products Co., whose principal products in peacetime were wheel rims and stainless steel beer barrels, is 99% on war work. Production of heavy-duty truck rims still constitutes an important part of the volume in this plant. However, the stainless steel barrel activity has been converted to the production of shatterproof oxygen cylinders for high-altitude flying, metallic belt links, bogie rollers, mooring anchors, motor frame bands fabricated from strip stock, rolled and welded rings, metal stampings.

Young Radiator Co., heat transfer specialists, have devoted their available facilities in the same direction, producing radiators, oil coolers, air coolers, and so forth, 100% for war requirements. Manufacturing procedures have remained the same. However, many of the facilities formerly used for the production of heating, cooling, and air conditioning equipment for commercial purposes have been converted to war needs.

In addition to the normal products made by Young, they have added such items as periscope holders, ammunition boxes, Navy spare parts boxes, ducts and other sheet metal parts for tanks, oil pans, frames, engine bases, and so on.

Federal-Mogul Corp. is 100% on war work, and has added several new plants in order to accommodate war activities such as airplane bearings and Navy propellers. Airplane bearing manufacture has entailed not only new plant facilities but new items of equipment such as diamond-boring machines and extensive facilities for lead and indium plating of silver bearings.

Federal-Mogul's marine division, formerly producing propellers for pleasure craft, has enlarged its facilities and is drawing upon outside subcontractors for the production of large propellers for the Navy and Coast Guard exclusively.

Fafnir Bearing Co. is 100% on war work, producing ball bearings. Plant and production facilities, including a number of entirely new plants, have been vastly increased to accommodate the demands of the war.

USL Battery Corp. reports that its total volume of storage battery production has been reduced by one-third due to elimination of civilian supply, despite the requirements of the Army and Navy.

■ Lamp Specialists Cooperate

Lamp specialists such as the Guide Lamp Division, General Motors Corp., and the Corcoran-Brown Lamp Division of Electric Auto-Lite Co. have made a major contribution in several ways. First of all, these manufacturers served on a special committee cooperating with the War Department on the development of blackout lighting equipment for military and combat vehicles. In addition, they cooperated in the Ordnance program for the development of the steel cartridge case.

The Thermoid Co. marks another type of activity which has contributed largely to the war program. Producer in peacetime of brake lining, clutch facings, radiator hose, fan belts, and universal joint discs, these same products are being made for the military services, with some proportion ear-marked for essential civilian needs.

Looms on which Thermoid formerly made carpet for passenger cars have been converted to weave duck for the Army. Cotton webbing is now made on some of the equipment formerly used for weaving asbestos tape. Much of the industrial hose equipment has been turned to making suction hose and water hose for the Navy and for the Office of Civilian Defense.

Thermoid's research and engineering talents now are directed to the improvement of the war products, to the elimination of critical materials, and to the development of new products urgently required in the war program. An example of this is the development of bulletproof rubber covered tanks of jettison types.

All of these activities have been accomplished by the conversion of existing equipment with but little plant expansion.

Houde Engineering Division, Houdaille-Hershey Corp., is typical of this progressive organization, being converted 100% to war work. This was accomplished without any reduction in personnel. The general character of the products includes such items as landing gear and hydraulic actuating cylinders, shimmy dampers, gun recoil mechanism, shock absorbing devices for military vehicles, and

many other products requiring precision work of high quality.

Carter Carburetor Corp., one of the leading carburetor producers in peacetime, is about 95% on war production, the remainder being accounted for in parts and service for essential civilian needs. This company found it necessary to just about double its existing facilities in meeting the demands of the French (before Pearl Harbor), for the British, and later for the U. S. military services. Among the products made by Carter are shell fuses, bomb fuses, carburetors, fuel filters, and an electric pusher fuel pump used for elimination of vapor lock in military vehicles.

Timken Roller Bearing Co., Bower Roller Bearing Co. and others are 100% on war production. Timken is currently producing tapered roller bearings, ball gun mount bearings, rock bits, alloy steel, and so forth. The burden of the war program has necessitated the addition of several new plants.

Stewart-Warner Corp. is devoting its energies 100% to the war, producing such new items as artillery and bomb fuses, instruments such as speedometers and gas gages, gasoline-operated heaters for airplanes. The normal automotive specialties are continued without change for use in tanks, trucks, buses, and in military vehicles. The changeover to war production was accomplished by converting existing facilities, with but little plant expansion.

Hoof Products Co., essentially a producer of governors, has swung over to war production about 90%. Latest development is a line of hydraulic valve parts and fittings for aircraft. This expansion in the line is being handled by a new building program now in progress.

■ Normal Products Abandoned

Woodall Industries, Inc. marks an example of an automotive producer whose normal products had to be abandoned in the interest of the war program. This company for 23 years specialized in the fabrication of fiberboard, die-cutting and pressing into shapes and assemblies for motor cars. The resulting experience in body engineering, equipment adaptation, die marking, and heavy-press operation built up a know-how which enabled the organization to swing over entirely to aircraft parts and assemblies including such parts as structural framing for military airplanes, engine cowlings, wing parts. This was accomplished by a conversion of existing facilities plus some added facilities which are under construction at the present time.

About two years ago, Woodall had members of the organization visit the airplane builders on the West Coast and then began experiments with the Guerin process, adapting it to the forming and cutting of airplane parts with inexpensive Masonite dies in its heavy presses. This work turned out to be quite successful, and has been further improved by additional developments by Woodall specialists in facilitating the formation of aluminum airplane structural parts.

Having this background, one might well question: "What economic gains can accrue from the unprecedented war program with its expenditures running into astronomical figures, and its resulting dislocations?" Fortunately, much can be salvaged from the waste of war. And much of the experience thus acquired can be turned to gainful use in the postwar automotive renascence.

For example, as a nation we have been prodigal in our use of natural resources. The war has not only taught the rules of conservation but, in the process, has brought to the fore the properties of useful materials not heretofore exploited. So far as materials go, we shall have more copper, more aluminum, more magnesium, more plastics, and more synthetic rubber than could have been conceived by the wildest stretch of the imagination.

We have learned that lower-priced steels can do the job formerly assigned to high-priced alloys. We have learned to do with less variety of specifications. After the war this may well lead to standardization that can simplify steel mill operation immeasurably.

We have learned other things. For example, the vast expansion of airplane production brought with it the headaches of an entirely new concept of surface finish, surface perfection, and dimensional tolerances. These were thought to be a definite bar to interchangeability and mass production techniques. But these difficulties were overcome with new types of machine tools, new techniques, new instrumentation and gages. After the war, many of these tricks can be turned to commercial production, with an improvement in the products to be made by the motor car industry.

Gear shaving has proved its virtues in airplane gear production. Precision thread grinding, formerly almost a laboratory process, is now a common procedure in most plants. And these threads can be cut directly from the solid bar. Precision grinding – to one or two microinches – is a production reality. Broaching has superseded rifling. There is virtually no end to the techniques that have been evolved and more or less painfully mastered in war production.

When the piston-ring producers were asked to finish the sides of rings for airplane engines to ten microinches, there was a cry of protest on all sides. Today, such rings can be finished in mass production – by the millions – to three microinches. The special equipment developed for the purpose will stand ready for use on automotive rings, to the benefit of engine life and performance.

Light weight was demanded of the high-capacity electric generators built for airplane installation. It constituted a new and major problem. Today, at Delco-Remy for example, generators of light-weight construction pack so much more output than do the largest bus or industrial equipment generators, in a smaller package and with such a reduction in weight, as to defy comparison. Is there any reason why this wartime lesson should be lost when the industry resumes its normal operations?

Equipment such as the Cincinnati Hydro-Tel, automatically producing intricately formed shapes and contours, was born of the war in its fullest expression of utilization. Literally hundreds of other machine tool developments came into being during the emergency. Most of these can be salvaged and utilized to the advantage of our postwar economy.

It is true that the industry's research and development program which normally goes on apace in the interest of progress was completely stalled due to the urgency of the war. But there has been a veritable flood of research endeavor throughout all industry. It has covered design, it has touched on machine shop practice, it has expanded our knowledge of cutting tools and cutting fluids, it has vastly increased our store of information on metallurgy.

The integration of all of these accomplishments will go far to make the post-war motor car a better product – more economical, longer-lived, and unquestionably lower priced.

Methods of STRESS DETERMINATION in ENGINE PARTS

by CHARLES LIPSON
Engineering Division, Chrysler Corp.

THE question of mechanical strength comprises one of the most essential factors in the design of machine and structural parts. Design must provide for a member strong enough to withstand the impressed loads and yet sufficiently light to conform to economic and functional requirements.

In the past, the conflict between high strength and low weight was usually resolved in favor of the strength considerations. This was particularly true if the engineering decision was being made against a background of past failures. Today this process cannot be tolerated, for the economic factors are more stringent and the design requirements far more demanding. The use of large doses of empirical formulas reinforced by factors of safety may have had full justification in the past. With the advent of high-speed, high-load machines this method of design became totally inadequate, although it still retains its usefulness for certain other applications.

The inadequacy of this practice arises from the fact that actual stresses in loaded members are often widely different from those anticipated by the ordinary laws of mechanics. In addition, there always existed uncertainty as to the operating loads and the properties of the materials used. This was recognized in the past by the introduction of the so-called "factors of safety." But even such an expedient was not totally satisfactory because the values were based on experience, and consequently their usefulness was strictly limited to the operating conditions from which they were evolved. In practical applications, this limitation was tacitly circumvented and high factors of safety were invoked to forestall possible failures.

The result was frequently an overdesign. For structural work in which considerations of safety exceed those of weight, such procedure may be fully justified. In airplanes and automobiles, however, weight saving is one of the essential requirements. It is in this field of engineering that the old methods of design prove inadequate and new methods, involving accurate measurements, are demanded.

This paper represents an attempt to describe these methods as used in Chrysler laboratories. Specifically it

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 12, 1943.]

THE old methods of structural and machine design, based on large doses of empirical formulas reinforced by factors of safety, are rapidly becoming outmoded in the fields of airplane and automobile engineering, due to the increasing importance of weight saving requirements, and new methods involving accurate stress measurements are being demanded, Mr. Lipson emphasizes.

The author describes four methods of experimental stress analysis that are used in the Chrysler laboratories, namely: photoelasticity, Stresscoat, extensometers, and electric strain gages. He also shows how they are applied in given engine problems involving evolution of a new design, determination of the cause of failure, and a comparison between various designs.

In the utilization of the four methods of experimental stress analysis described, there are two principal means of procedure: static tests, using simulated service loading, and dynamic tests, under actual service conditions.

Static load testing, Mr. Lipson says, is usually the most convenient and generally favored because the problem of instrumentation is greatly simplified. The penalty for this simplification lies in the fact that the investigation is often conducted on the basis of loads that may or may not correspond to operating conditions. Dynamic load testing demands more complex instrumentation and so far it is less thoroughly developed. A comprehensive stress analysis investigation requires both phases of testing: dynamic tests should establish the mode and magnitude of operating loads, while static measurements will determine the corresponding stresses. Refinements on instrumentation and technique are necessary to promote greater accuracy and speed but this particular phase of experimental stress analysis is developing in a satisfactory manner.

★ ★ ★

THE AUTHOR: CHARLES LIPSON has been with the Engineering Division of the Chrysler Corp. since 1936, specializing in mechanical development and research, particularly in the field of experimental stress analysis and life testing. Formerly an instructor in physics at the College of Arts and the School of Engineering, New York University, he has presented several papers before the American Physical Society. Mr. Lipson was graduated from Muhlenberg College in 1930 with a B.Sc. degree, and from New York University in 1935 with a Ph.D.

involves the application of stress measurements to the development of engine parts.

I—EXPERIMENTAL METHODS OF STRESS ANALYSIS

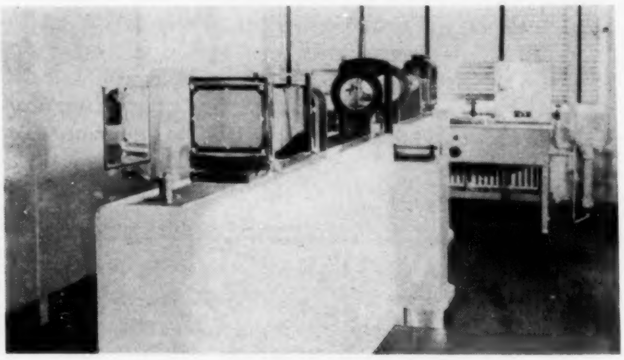
■ Photoelasticity

Photoelasticity may be defined as the science of measuring stresses in an elastic body by optical means. A polarized beam, upon entering a stressed transparent plastic model, will divide into two components traveling in the two principal planes. Due to the difference in velocity of these two components, optical interference takes place and the fringe number becomes an index of acting stress according to the relation:

$$P - Q = \frac{kn}{t}$$

where P and Q = principal stresses
k = fringe constant of the material
n = number of fringes
t = thickness of plastic model

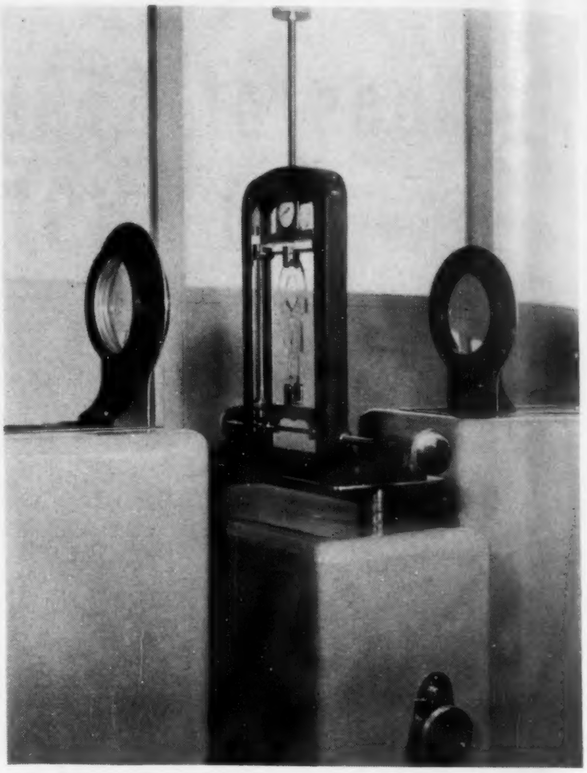
The polariscope used in the Chrysler laboratories for this work is shown in Fig. 1. The polarizing-analyzing prisms are of the Glan-Thompson type; the field of view is 7 in., filtered mercury light being used as a source. Quarter-wave plates are incorporated in this instrument.



■ Fig. 1 – Polariscope used in Chrysler laboratories

A universal loading frame placed in the field of the collimated beam is shown in Fig. 2. The plastic model is held at the base by means of a suitable bracket and at the top by a clamp which fits around a calibrated loading beam. Load is applied by means of two lead screws turned by a side knob through a set of bevel gears. The knob mounted on the opposite side serves to adjust the frame laterally, the one on the top vertically, so that the loaded plastic can be centered in the field of the polariscope. An additional coarse vertical adjustment for the entire loading frame has been provided as shown.

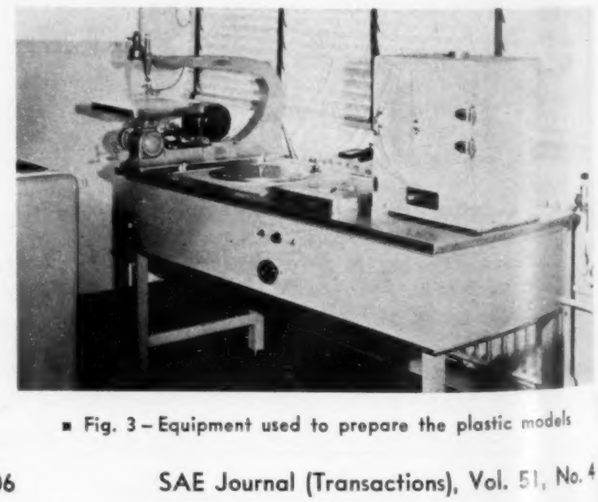
It is imperative that plastic specimens be prepared with utmost care; and Fig. 3 shows a convenient means of doing this. The plastic is first polished (if it is obtained in an unpolished form) on the 12-in. polishing machine, and then annealed in the oven. Once the metal template has been machined, its outline is traced on the plastic. The outline is first cut roughly on a scroll saw, and then finely finish-machined on the vertical mill.



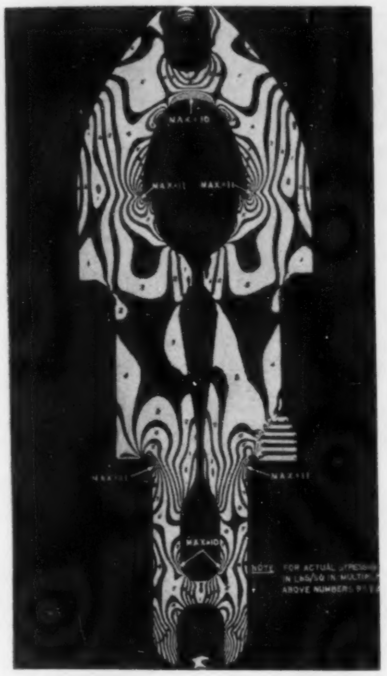
■ Fig. 2 – A plastic model in a universal loading frame placed in the field of the collimated beam

The problem of a shaft coupling (Figs. 2, 4, and 5, longitudinal-center-section view shown) represents a simple example of the application of photoelasticity to design work. The object of the test was to determine the difference between an undercut fillet, as it is shown on the left side, and the same section reinforced by fins, as shown at right. From the fringe pattern it appears that fins did not reinforce the fillet section.

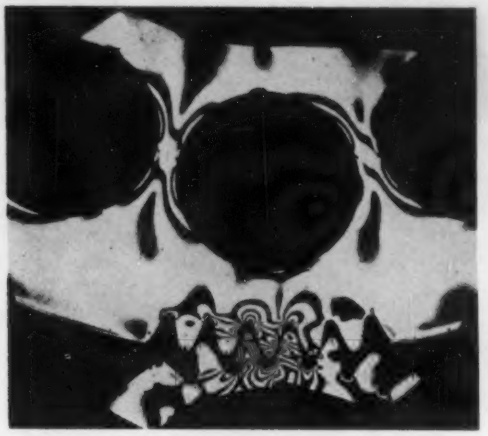
A second example of photoelastic work involved stress analysis of meshing gears. The object of this test was to determine the effect produced by lightening holes cut in the gear web, as shown in Fig. 6. From the count of the stress fringes and a calculation of the force at the section studied, it was concluded that the stresses produced by the lightening holes are not excessive.



■ Fig. 3 – Equipment used to prepare the plastic models

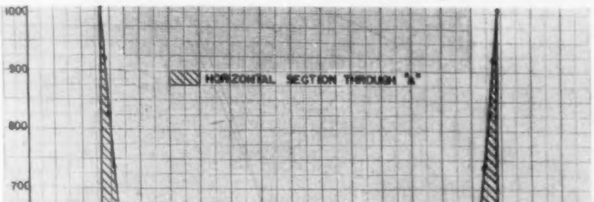


■ Fig. 4 – Photoelastic view of a shaft coupling – longitudinal center-section



■ Fig. 6 – Photoelastic view of meshing gears – effect of lightening holes

■ Fig. 7 – Photoelastic view of a strap consisting of a straight section and a round section



■ Fig. 5 – Stress distribution diagrams for the shaft coupling

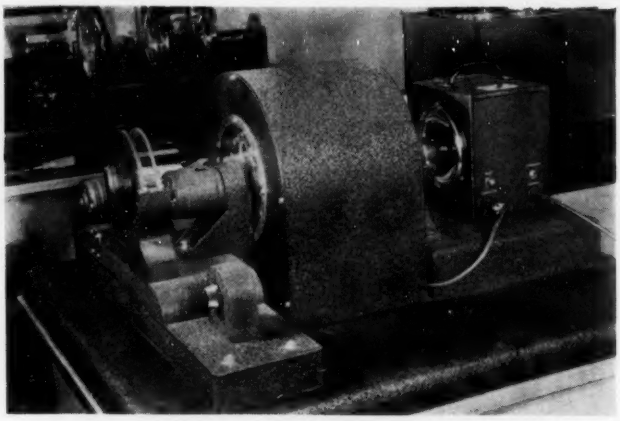
Still another problem (Fig. 7) involved determination of the improvement which could be anticipated when a straight-section strap was substituted for one of round section. Photoelastic data revealed nearly an 80% improvement.

Centrifugal Stresses – For study of stresses resulting from centrifugal forces, two experimental methods are available.

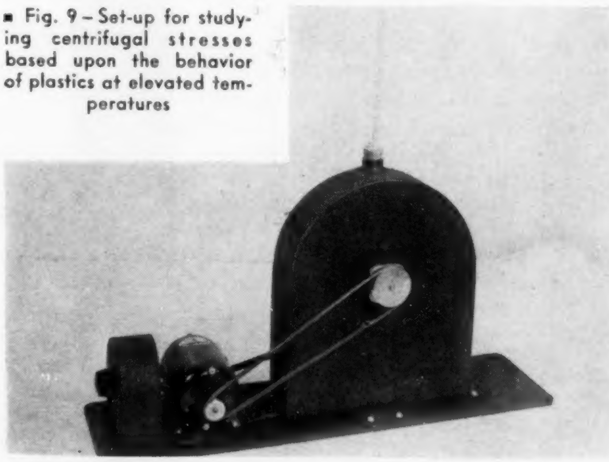
The first method consists of setting the rotating plastic model in the field of a conventional polariscope, but instead of a standard light a stroboscopic source is used.

The laboratory fixture used for this purpose is shown in Fig. 8. The plastic is mounted on the shaft of a small motor and rotated under a protective cover to speeds up to 10,000 rpm. A commutator on the motor shaft is connected to the contactor of the strobotac, producing one flash of light per revolution. Stress fringes are viewed through the analyzer shown on the left; stresses are then evaluated by conventional photoelastic means.

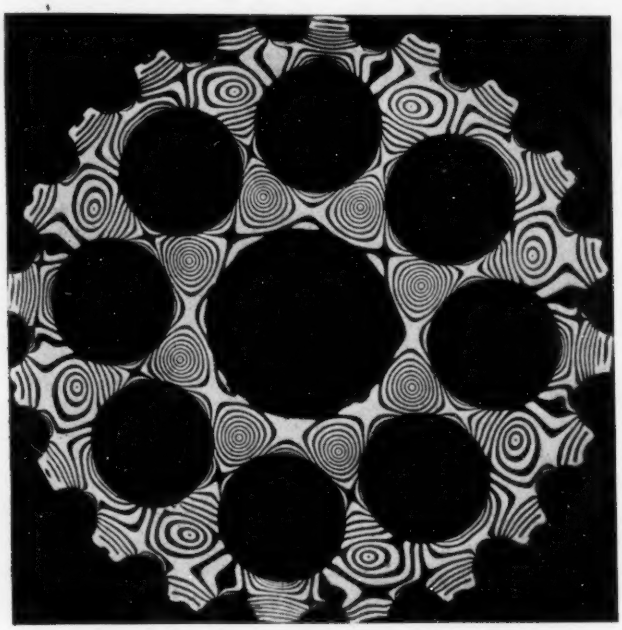
The principle of the second method is based upon the behavior of plastics at elevated temperatures; and its successful development is largely due to M. Hetényi of the Westinghouse Research Laboratories. Above a certain temperature, the plastic model yields, and the external force impressed upon it will be relieved by the deflection of the material. On subsequent cooling, the model will retain



▪ Fig. 8 – Set-up for studying centrifugal stresses using a conventional polariscope



▪ Fig. 9 – Set-up for studying centrifugal stresses based upon the behavior of plastics at elevated temperatures



▪ Fig. 10 – Effect of lightening holes on the stress distribution in a high-speed rotor

the impressed deflections and will exhibit the resultant stress pattern.

Fig. 9 shows a fixture used in Chrysler laboratories for one of such problems. The plastic rotor is mounted on a shaft inside an electric oven, rotor speed being regulated by means of a rheostat. To obtain the fringe pattern shown in Fig. 10, the plastic rotor was spun at 2300 rpm through the following temperatures:

From 80 to 180 F for ½ hr
At 180 F for 1 hr
From 180 to 80 F for 3 hr

The object of this investigation was to determine the effect of lightening holes on the stress distribution in a high-speed rotor. From the fringe pattern obtained it was determined that:

1. Maximum stress occurs at the edges of the lightening hole.

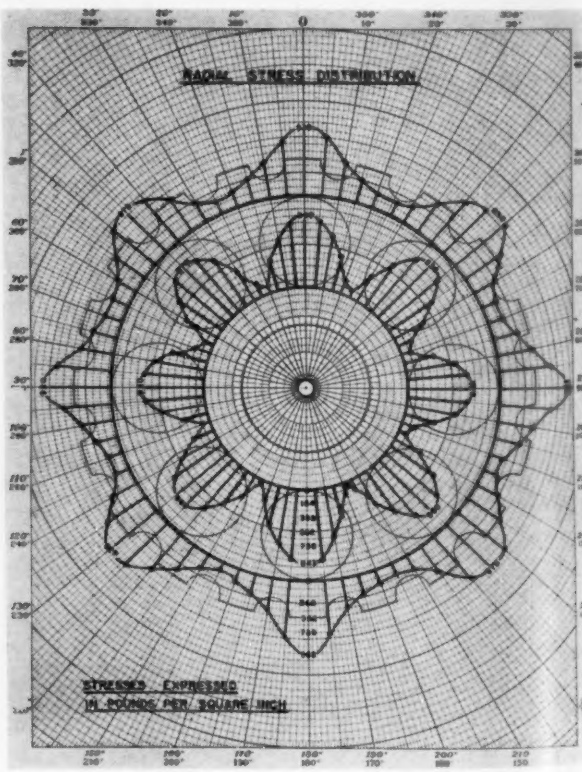
2. Stresses at the edge of the outer groove are 30% lower.

Stress distribution in the rotor is shown in Fig. 11.

In spite of the successful application of the photoelastic method to a large variety of problems, the technique is not suitable for numerous other problems in which the third dimension comprises an essential requirement of design. For those problems a three-dimensional photoelastic technique was developed, and although more difficult and time-consuming it represents a significant step in the development of a useful tool for experimental stress work.

Three-Dimensional Stresses – There are at present two methods of three-dimensional analysis:

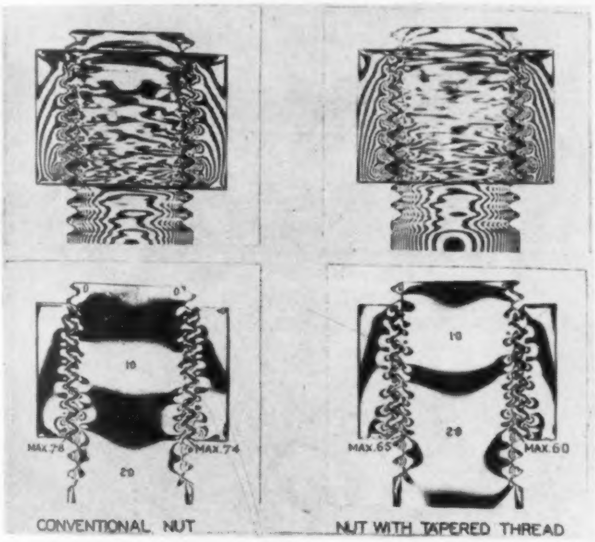
(a) Freezing method
(b) Light-scattering method



▪ Fig. 11 – Radial stress distribution in the high-speed rotor

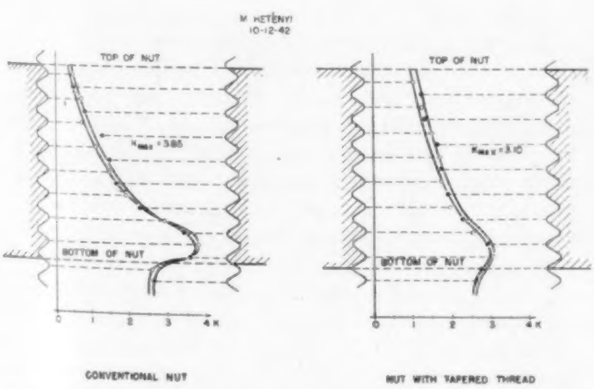
The technique of the freezing method is similar to that used in two-dimensional photoelasticity in that the analysis is made on thin plastic samples sliced from the three-dimensional plastic model under consideration. The model is loaded, placed in the oven, and "frozen." Slices are carefully cut and polished and then analyzed in the field of a conventional two-dimensional polariscope.

An excellent example of a problem successfully analyzed by this method is offered by M. Hetényi. The problem involved study of stress distribution in bolts for various nut designs. Fig. 12 shows the fringe pattern for a nut with a conventional thread compared with a tapered thread. The upper photographs are of the three-dimensional model before slicing and thus offer only a qualitative picture of the stress distribution. The bottom photographs show thin slices of the center section, carefully cut and polished. It is from these slices that the stresses were evaluated quantitatively.



■ Fig. 12 – Fringe patterns for a conventional nut and a tapered-thread nut

Fig. 13 represents a graphical plot obtained from the foregoing stress photograph. The graph shows that the maximum stress occurring in a conventional nut and bolt fastening can be reduced by about 20% through the application of nuts with a tapered thread.

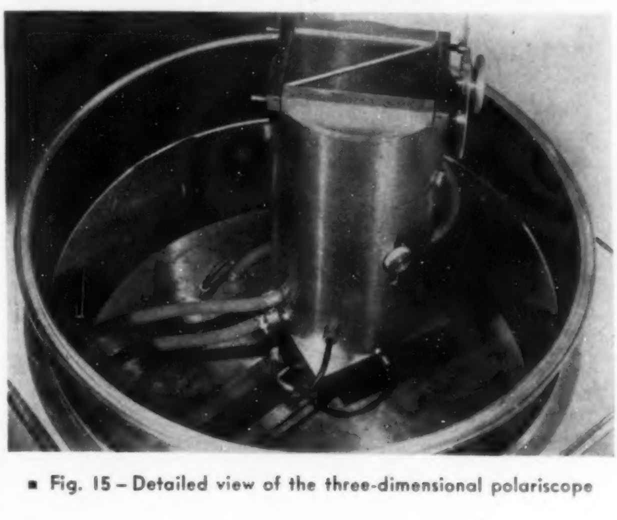


■ Fig. 13 – Stress distribution in the threads of a conventional nut and a tapered-thread nut

The light-scattering method, developed by R. Weller of Washington State College, is based on the principle that when light is scattered within a material medium at right angles to the incident beam, the scattered light is plane polarized. This scattering property is used as an analyzer in a polariscope, while the polarizer is of conventional construction.

The three-dimensional polariscope used in Chrysler laboratories is shown in Figs. 14 and 15. Its constructoin is based on the optical system developed by Weller. Light coming from a 1000-w mercury arc lamp is collimated by two 6-in. condensing lenses, passed through a 6-in. polaroid, and then through an adjustable slit. The construction is flexible, allowing for the rotation of the base, horizontal adjustments on the large dovetail track, and fine adjustment of the square base. The upper drum supports the camera and also the three-dimensional plastic model in its loading frame. The latter is placed in a Halowax oil solution, whose index of refraction is the same as that of the plastic model. The purpose of this is to compensate for the

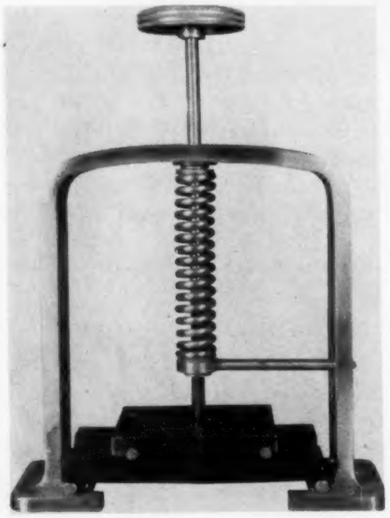
■ Fig. 14 – Three-dimensional polariscope based on light-scattering method



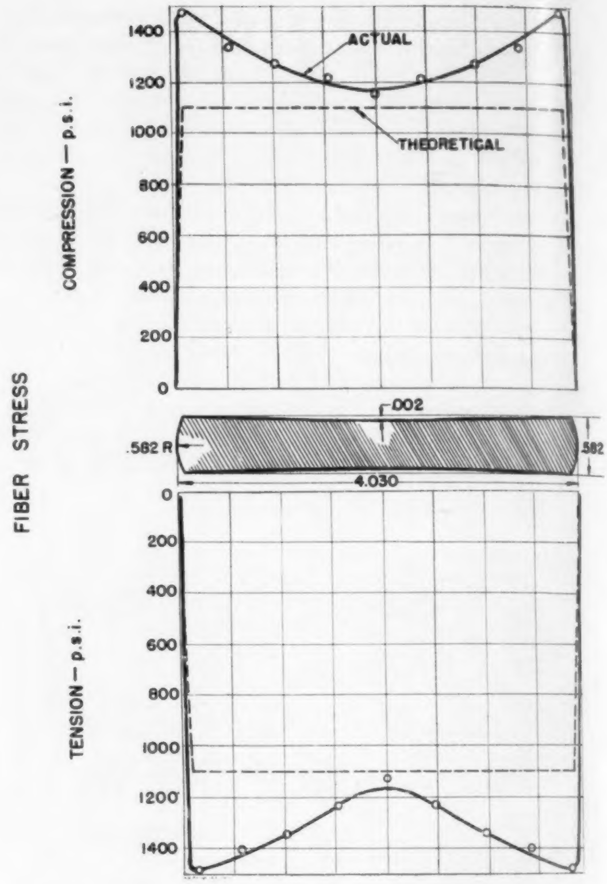
■ Fig. 15 – Detailed view of the three-dimensional polariscope

uneven path of the scattered light, resulting from the curvature of the plastic model.

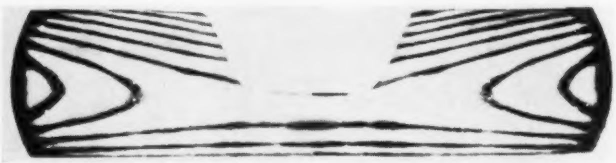
Using this method of analysis, the location of the neutral axis and the magnitude of maximum stresses were determined for four different designs of an experimental segment. The loading frame, with a model of one of these sections, is shown in Fig. 16. The photoelastic stress pattern of the model itself is shown in Fig. 17. The experimentally determined stress data are given in Figs. 18, 19, 20 and 21.



■ Fig. 16 – Loading frame and experimental model for use with the three-dimensional polariscope



■ Fig. 18 – Stress distribution for the flat section



■ Fig. 17 – Photoelastic stress pattern of the grooved section

As to the application of the photoelastic stress to actual practice, stress values obtained from a plastic model can be directly applied to its metal prototype, because theoretically stress distribution is solely a function of the geometry of the part. The validity of such a correspondence has been verified on numerous occasions by other methods of experimental stress analysis such as extensometers, Stresscoat, and electric strain gages. However, since all the considerations involved are not yet fully understood, a certain restraint should be exercised when this application is being made.

The principal advantages of photoelasticity can be classified as follows:

(a) A complete view of the stress pattern can be obtained.

(b) Stresses in otherwise inaccessible regions (for example, fitted members) can be studied.

(c) The test can be conducted before an actual part is made.

(d) The technique is fast and inexpensive.

The principal disadvantages are:

(a) Two-dimensional models do not always represent sections of maximum stress, especially when combined stresses (for example, torsion and bending) are encountered.

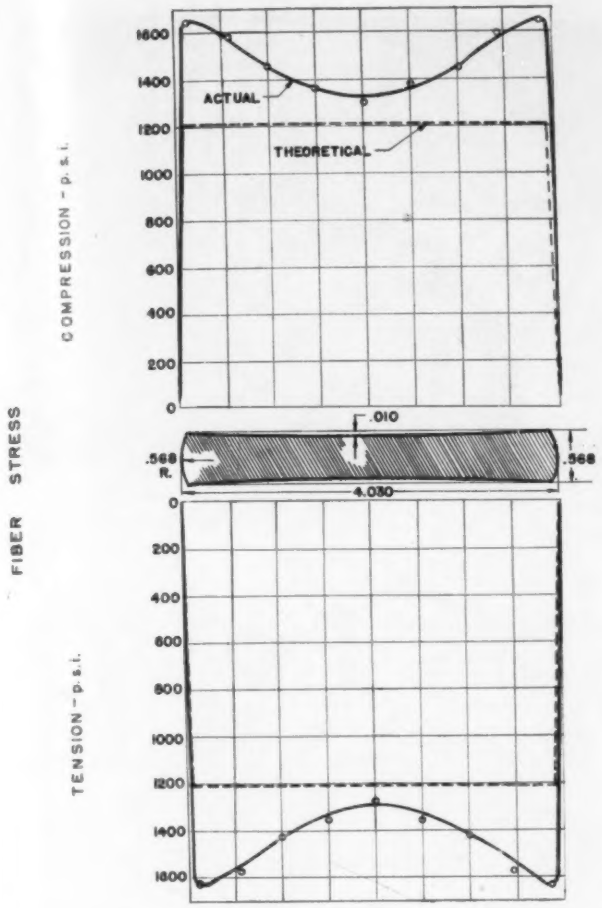
(b) Three-dimensional technique is difficult and time consuming.

(c) As yet, dynamic stresses, especially impact, cannot be studied too successfully by means of photoelasticity.

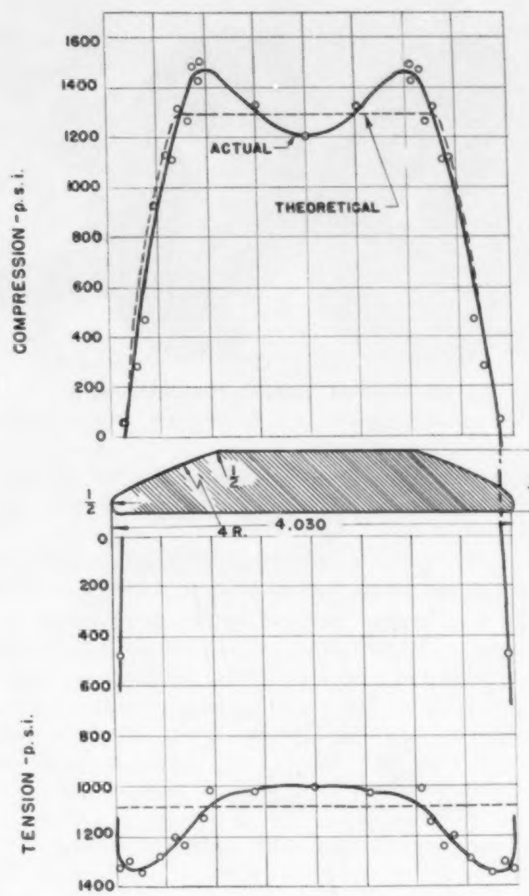
■ Stresscoat

Stresscoat is a brittle lacquer which fractures at low values of tension strain (0.0005 to 0.001 in. per in.). Because of this property, it may be used as a coating on loaded parts to indicate the location, direction, and magnitude of tension strain. Compression strains may also be measured by allowing the coating to come to a neutral condition while the part is under maximum load. Stresscoat is commercially available through the Magnaflux Corp.

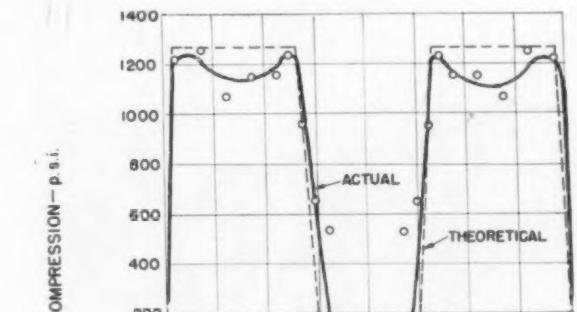
A part to be tested by the Stresscoat method is first thoroughly cleaned, then sprayed with an aluminum undercoat to provide a reflecting surface which makes the stress pattern more easily visible. Stresscoat is then applied by means of an airbrush in an even coating 0.003 to 0.005 in. thick. Since the sensitivity of the lacquer is affected by temperature and humidity, it is made in several grades for varying conditions. The proper grade of lacquer for each test is chosen from a selection chart, according to readings secured



■ Fig. 19 – Stress distribution for the concave section



■ Fig. 20 – Stress distribution for the parabolic section



■ Fig. 21 – Stress distribution for the grooved section

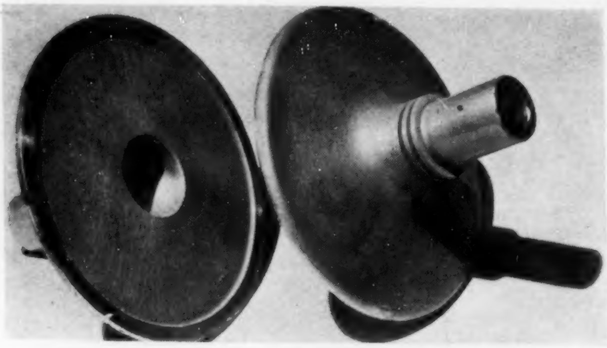
with a sling psychrometer in the place where the test is to be run.

Calibration strips are sprayed at the same time and with the same lacquer as the part to be tested. The strips and the test piece are then dried together for at least 6, preferably 12, hr. Drying overnight is generally the most practical period. To secure a quantitative calibration when the test is made, the strip is placed in a cantilever-beam fixture and given a known deflection by means of a cam.

A reasonable load is then applied to the test piece, the part is examined for stress cracks, and then the load is removed. After a normalizing period of twice the total time of the first cycle of load, a new load 20% higher is applied, the piece re-examined, and the process repeated.

Stress investigation of assembly end plates illustrates the application of Stresscoat. This test was conducted to determine the tensile and compressive static stresses in the end plates of an assembly which was held together by a bolt through its longitudinal axis. The end plates were loaded axially by means of a suitable fixture in an Olsen testing machine.

The resultant stress pattern is shown in Fig. 22. The member on the left was placed to show the tensile stresses, the one on the right, the compressive. The highest tensile stress was found to be in the sharp corner at the base of the counterbore. This stress was 95,000 psi and was above the yield strength of the material (85,000 psi). Yielding at this point, however, was highly localized and should be of no



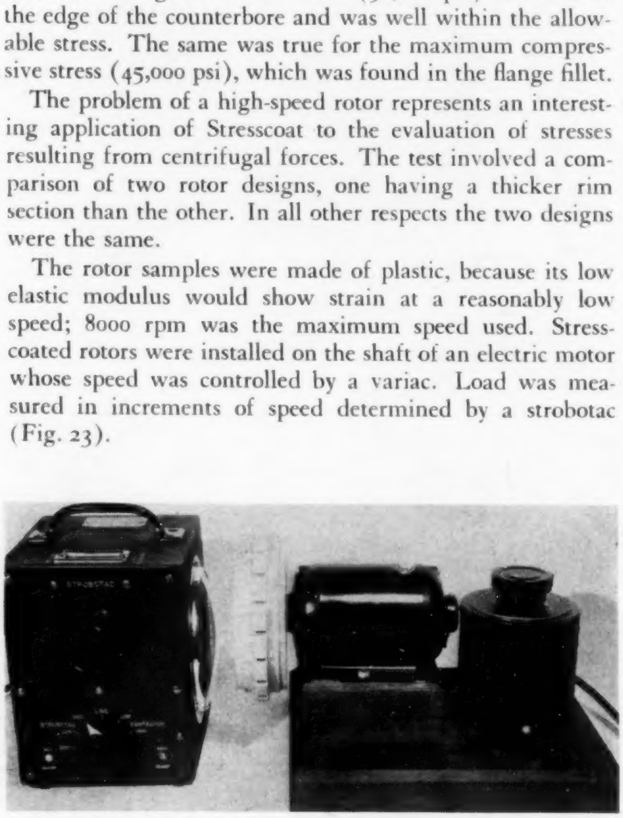
■ Fig. 22 – Stresscoat pattern of assembly end plates

consideration for static loading. If any extraneous vibration is expected, the counterbore must be filleted in order to avoid failure.

The next highest tensile stress (50,000 psi) occurred at the edge of the counterbore and was well within the allowable stress. The same was true for the maximum compressive stress (45,000 psi), which was found in the flange fillet.

The problem of a high-speed rotor represents an interesting application of Stresscoat to the evaluation of stresses resulting from centrifugal forces. The test involved a comparison of two rotor designs, one having a thicker rim section than the other. In all other respects the two designs were the same.

The rotor samples were made of plastic, because its low elastic modulus would show strain at a reasonably low speed; 8000 rpm was the maximum speed used. Stresscoated rotors were installed on the shaft of an electric motor whose speed was controlled by a variac. Load was measured in increments of speed determined by a strobotac (Fig. 23).



■ Fig. 23 – Set-up for determining the effect of centrifugal force on a high-speed rotor – Stresscoat method

The general stress pattern (Fig. 24) for both designs was found to be the same. The highest stress in each (Fig. 25) was radial, at the fillet joining the hub and the web. The tangential stresses in the rim were found to be higher on the inner surface than on the outer. Between the hub and the rim a biaxial pattern, caused by the combination of radial and tangential stresses, was observed. Of the two designs tested, the one with a thin rim (20% thinner than the other) exhibited 20 to 25% lower stresses.

The principal advantages of Stresscoat can be listed as:

(a) Of all the experimental means of stress measurement, Stresscoat represents the fastest method for obtaining a complete stress pattern of a loaded member.

(b) The effective gage length is extremely small.

(c) Stresscoat may be used effectively for the determination of stresses in a rotating part.

(d) Stresscoat checks may be obtained under dynamic loading conditions.

(e) Stresscoat can be used as a preliminary step in stress analysis, to be followed by more accurate experimental means (extensometers, strain gages).

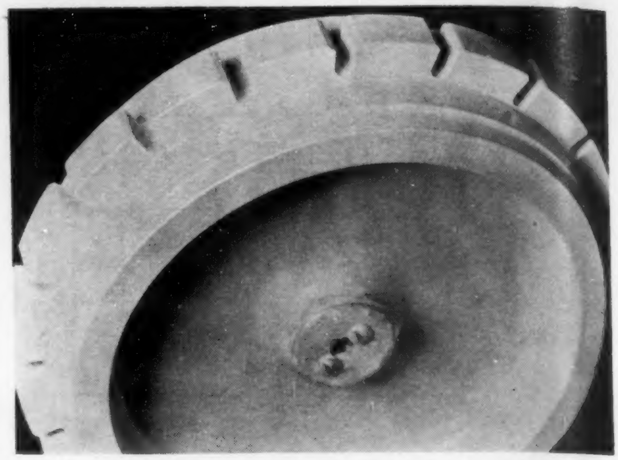
The principal disadvantages are:

(a) Stresscoat is sensitive to temperature and humidity fluctuations, and unless used in a controlled atmosphere it produces a craze pattern.

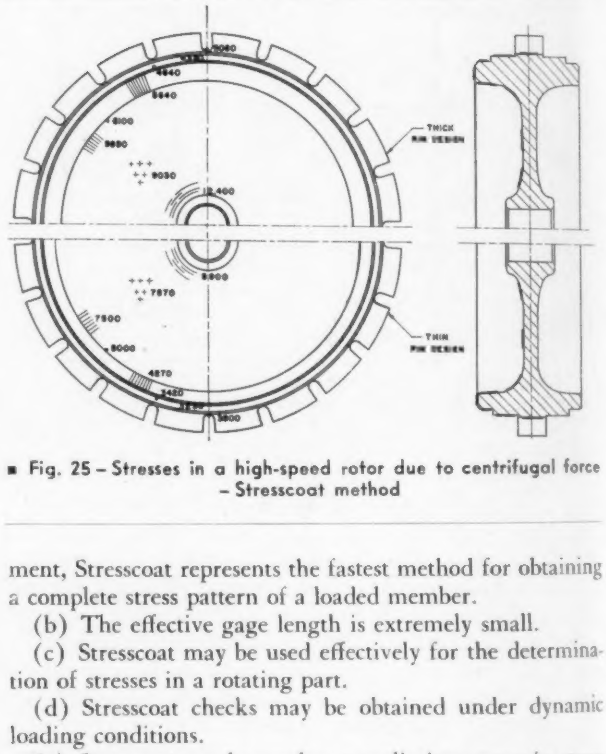
(b) Thickness of lacquer affects its sensitivity, and consequently the quantitative accuracy of Stresscoat depends largely on the skill and experience of the operator.

(c) The behavior of Stresscoat under biaxial strain is uncertain.

(d) Quantitative accuracy of Stresscoat is low, although it can be enhanced by taking an average of two or three careful checks.



■ Fig. 24 – Stresscoat pattern of a high-speed rotor



■ Fig. 25 – Stresses in a high-speed rotor due to centrifugal force – Stresscoat method

■ Extensometers

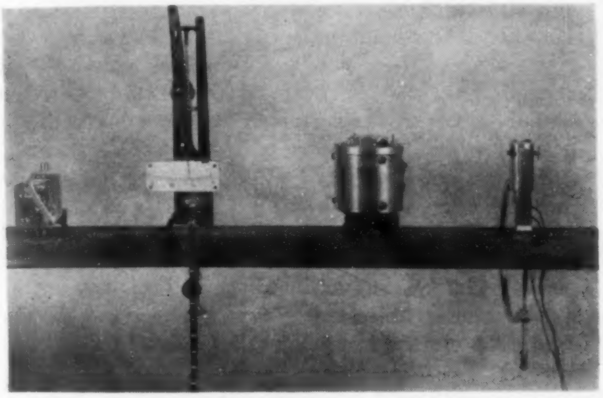
Fundamentally, an extensometer is a device for accurately measuring changes in distance between two fixed

points on a part subjected to an external load. The distance between the two points, previous to the load application, is known as the gage length. Depending on the type and use of the extensometer, the gage length may be several inches or a small fraction of an inch. For stress analysis work only short gage extensometers are of practical importance.

Such extensometers require an elaborate provision for accurate magnification of the actual strain. This is accomplished by mechanical, optical, and electrical means, or their combination. Different extensometers are characterized by different methods of magnification.

There are six main types of extensometers in use today, some of which, used in Chrysler laboratories, are shown in Fig. 26.

a. Huggenberger
b. Tuckerman
c. Tensor
d. Maze
e. Photoelectric
f. Electromagnetic



■ **Fig. 26 – Extensometers (left to right, A, B, C, D) used in the Chrysler laboratories**

a. *Huggenberger* – This instrument (*B* in Fig. 26) is dependent only on mechanical magnification. Strain readings can be made to 0.00001 in. in a 1-in. gage length. The shortest gage length is ½ in., and the magnification ratios vary from 2000 to 3000 times. This instrument is one of the oldest and most commonly used. Its chief disadvantages are its weight, size, and excessively long gage length.

b. *Tuckerman* – This gage is based on the principle of the optical lever system, readings being taken by means of an auto-collimator. The sensitivity of this gage is high (0.000002 in.). The shortest gage length commercially available is 1 in., although it may be possible ultimately to measure strain conveniently and accurately on a ¼-in. gage length. Because it involves the use of a collimator, it is not convenient for rapid measurement of stress.

c. *Tensor* – This instrument (*C* in Fig. 26) contains three legs 120 deg apart, and thus the direction as well as the magnitude of the principal stresses can be determined without recourse to any other method. The combined optical and mechanical magnification is 400 times when viewed through a six-power magnifying glass. The distance between gage points is approximately 1 in., and therefore this type is more applicable to structures containing large uniform surfaces.

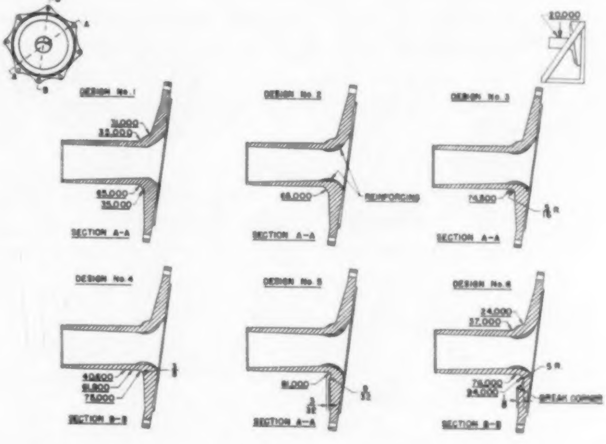
d. *Maze* – The Maze gage (*A* in Fig. 26) construction is based on the combined mechanical and optical principle, the reflected light being viewed through a telescope. It is a short-gage-length instrument, being commercially available in sizes from 0.2 to 0.6 in. Because it involves the use of a telescope it suffers from the same disadvantages as the Tuckerman gage.

e. *Photoelectric* – Magnification in this instrument (*D* in Fig. 26) is achieved mechanically, optically, and electrically. Operation of the gage is through a single lever which shifts one of two 120-lines-per-inch gratings. These in turn intercept a collimated beam of light and direct it on a photoelectric cell. The current generated by the cell is read directly on a microammeter without intermediate magnification. The overall magnification is 30,000 times. This instrument has been successfully developed by the General Motors Research Laboratories Division and is commercially available in gage lengths of ¼, ⅛, and 1/16 in. The instrument is small and light, and it is eminently satisfactory for general stress analysis. Success in using it, however, demands care and manual dexterity.

f. *Electromagnetic* – The function of this instrument depends upon inductive changes in two coils separated by a diaphragm. The latter is connected to a movable pivot so that its motion, transferred by a link and pin arrangement, changes the air gap between the two coils. Instruments of this type have been constructed in Europe with a gage length as short as 0.5 mm.

Study of the motor support trunnion represents the type of problem which is most adequately approached by a short-gage-length extensometer technique.

Stress investigation was conducted on six different modifications of the standard trunnion by means of Stresscoat and the 1/16-in. photoelectric extensometer. The trunnion was bolted to a vertical face plate, placed in the Olsen machine and loaded along the direction of the motor mounting centerline. The design forms studied and the resultant stress values are given in Fig. 27. Apparently, none of the modifications tested has produced improvement over the trunnion originally designed (design No. 1).



■ **Fig. 27 – Stress analysis of motor mounting trunnion – measurements taken with Stresscoat and with photoelectric extensometer**

In this application, the undercut fillet does not seem to offer any advantage over the standard fillet.

The trunnion of the original design was characterized by uniform stresses in all sections measured except one (65,000 psi), for which the stress was over 90% higher than in the remaining sections. It could not be readily ascertained at the time whether this could be relieved by a plastic flow, and consequently it was found necessary to conduct the present series of tests. The principal advantage derived from this investigation was the fact that it eliminated fillet contour modification as a solution for the difficulty encountered in the original design.

The principal advantages of extensometers can be listed as:

(a) They may be designed to measure strain over a very short gage length.

(b) The method has a high degree of accuracy.

(c) Strains at a point may be measured quickly in more than one direction.

The principal disadvantages are:

(a) Extensometers are limited to static work only.

(b) Although some extensometers are relatively small, they are still too large to measure strain in small sections (for example, ⅛-in. fillets).

(c) Proper use of small-gage-length extensometers demands considerable patience and care.

■ Electric Strain Gages

Electric strain gages are gages in which a variation in average strain is accompanied by a proportional change in their electrical characteristics. This is then measured with suitable electrical instruments.

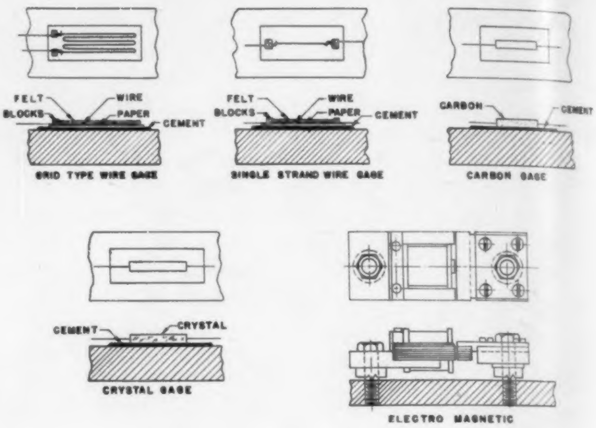
The electrical impedance of the gage is the customary variable used. This means that the changes to be measured are those of resistance, inductance, capacitance, or any combination of these. Since the measurement of impedance can be made very accurately with instruments of high sensitivity, it follows that the change in average strain which can be detected by an electric gage is very minute. Another electrical characteristic which can be used is the piezoelectric effect exhibited by certain crystals. That is, if a voltage is applied across the faces of a piezoelectric crystal, the crystal will distort mechanically. Conversely, a crystal mechanically distorted exhibits a potential difference across its faces. When crystals are used as strain gages the mechanical distortion creates voltage proportional to the strain.

There are four main types of electric strain gages in use today (Fig. 28):

a. Electromagnetic
b. Piezoelectric
c. Carbon Resistance
d. Wire Resistance

a. *Electromagnetic* – The reluctance of the magnetic circuit is varied by changing the length of the air gap. This produces a varying inductance which can be measured by an impedance bridge, or some other device. The measurement may be static or dynamic.

The gage fastened to the test piece with bolts is usually 1 or 2 in. long. This limits the use of this type of gage to an investigation of large parts. However, this type of gage is very sensitive, reliable, and involves a minimum of auxiliary equipment.



■ Fig. 28 – Types of strain gages

b. *Piezoelectric* – These gages are approximately ½ in. long and are fastened to the test piece with cement. When a strain is imposed, voltage difference appears across the crystal, and it can be measured when suitably amplified. Usually, only dynamic strains are measured with these gages.

The main advantage of a piezoelectric gage is that a very large signal is produced by a relatively small strain. The disadvantage lies in the fact that they cannot conform to curved surfaces. In addition, when compensated for temperature changes, the frequency response is not flat. The gage is more useful for frequency and relative amplitude determination than for quantitative strain measurements.

c. *Carbon Resistance* – The resistance of these gages increases under tension and decreases under compression. They are commercially available and can easily be made by filing a carbon resistor flat and then cementing it on the tested part. The signal is obtained either from a bridge or by measuring the change in voltage across the gage. Since atmospheric conditions affect these gages they are usually limited to qualitative data under dynamic loading conditions, though quantitative results have also been reported.

d. *Wire Strain Gages* – These gages are eminently satisfactory for the great majority of engineering applications, and as a result they are widely used in various laboratories and industrial organizations. Among others, several aircraft firms utilize them for the study of stresses in the aircraft structure, not only under static conditions, but also in flight.

As in the case of carbon gages, the resistance of the wire gages increases with tension and decreases with compression. The signal is obtained in an analogous manner and the method of attachment similarly involves the use of cement. Atmospheric conditions do not materially affect these gages; and, if necessary, temperature compensation may be made without affecting the linearity of the signal. Qualitative as well as quantitative strain readings can be obtained both statically and dynamically up to temperatures as high as 400 F. These gages have been made with a gage length as short as ¼ in. and even ⅛-in. gages will soon be commercially available.

A satisfactory static strain measuring instrument avail-

able is the Baldwin-Southwark SR-4 bridge with a null indicator galvanometer. The instrument has a high sensitivity but it is adversely affected by mechanical vibrations. The Baldwin-Southwark Portable Bridge has an electronic null indicator and thus is very much less susceptible to mechanical disturbances. At the same time its sensitivity is lower, and therefore its application is limited to gages of at least 100-ohms resistance. There is also commercially available a strain gage recorder and an automatic switching panel, by means of which strain gage records of 48 gages can be taken in a short interval of time.

For measurement of dynamic stresses by means of strain gages, instruments commercially available are cathode ray oscilloscopes and multi-element galvanometer-type recording oscillographs.

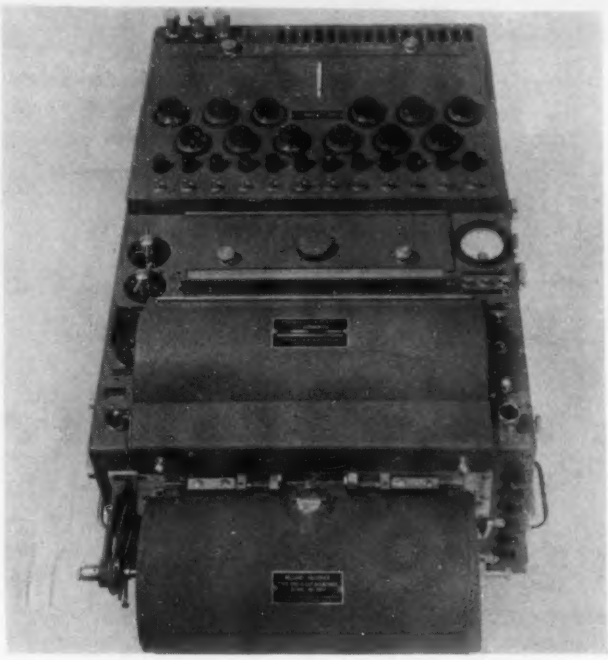
The cathode ray oscilloscope is an ideal instrument for dynamic strain work when only one gage at a time is being measured. It indicates wave shape and peak values which cannot be read by means of a meter. Usually these two observations are more important for an analysis of a loaded member than any meter reading could be. Cathode ray oscilloscopes may be made so that the frequency response is flat from 1 to 30,000 cps. If a response down to d-c is required, a d-c amplifier may be connected directly to the deflecting plates of the oscilloscope. Another advantage of this instrument is that voltage amplifiers only are needed. To obtain a permanent record, the cathode ray screen may be photographed.

The cathode ray oscilloscope is an essential tool in any kind of a dynamic measurement, and consequently it has found a widespread application in all fields of science and engineering. Several of these instruments are available in Chrysler laboratories; one (9-in. screen) is shown in Fig. 29.

In studying phase relationship between strains, the cathode ray tube requires special electronic equipment or a special multibeam tube. Therefore a more convenient means of obtaining a simultaneous record of several strain gages is to use a multi-element galvanometer oscillograph. Two of the several different oscillographs available have been found particularly suitable to our needs:

(a) The Hathaway Oscillograph, Type S5, (Fig. 30) is a 12-channel instrument, each galvanometer element having a natural frequency of 3000 cps and a minimum sensitivity of 4 mm per ma. The record may be taken on film 6 in. wide and up to 200 ft long, or on sensitized paper 6 or 10 in. wide and also up to 200 ft long.

■ Fig. 29 – A cathode ray oscilloscope



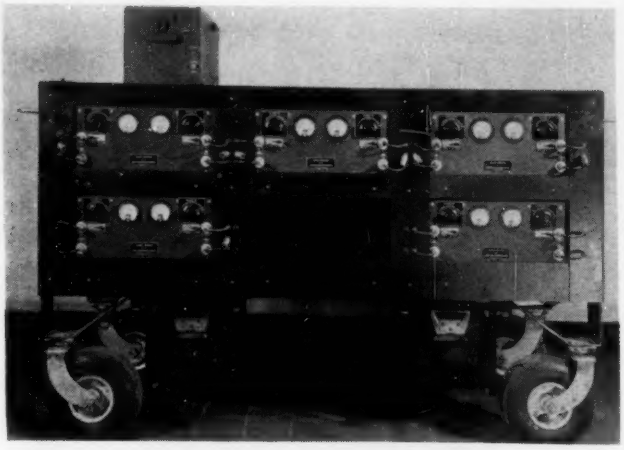
■ Fig. 30 – Hathaway oscillograph

(b) The Miller Oscillograph, type E-12, has 12 elements, 10 of which have a natural frequency of 2000 cps with a sensitivity of 150 ma rms per in. peak to peak. The other two elements have a sensitivity of 15 ma rms with a correspondingly lower natural frequency. The record may be taken on film or sensitized paper which is 6 in. wide and up to 200 ft long.

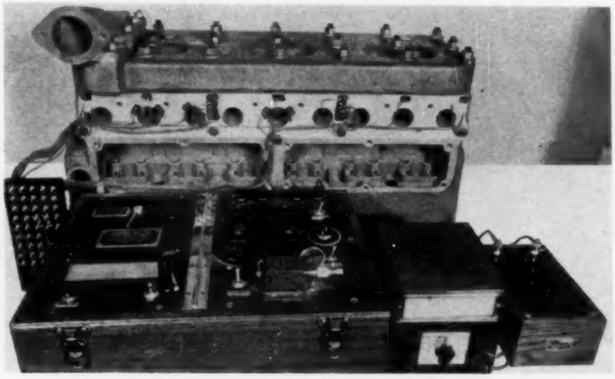
This instrument together with its amplifiers and power supply is shown in Fig. 31.

Fig. 32 shows an application of electrical strain gages to static stress analysis work. The purpose of this investigation was to compare two cylinder-block designs as a function of the cylinder-head-stud tightening. The machined face of the block was Stresscoated, load applied by tightening the studs, and the stress pattern obtained. Several strain gages were then applied to the high stress regions thus located. The resultant data have shown that as a result of a water jacket change, stresses in the regions of failure decreased by 30%. This study also involved comparison between various orders and arrangements of bolt tightening.

The valve rocker arm represents the application of strain gages to dynamic work. The purpose of this investigation was to determine operating loads impressed on a valve



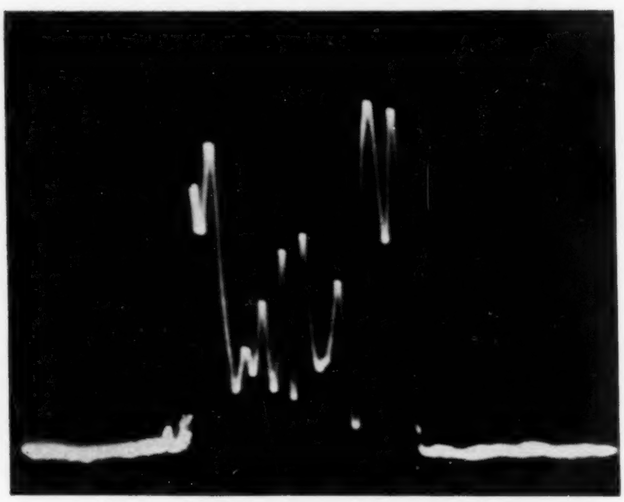
■ Fig. 31 – Miller oscillograph with amplifiers and power supply



■ Fig. 32 – Application of electrical strain gages to an investigation of a cylinder block

rocker arm in service. This was accomplished by installing wire gages on the rocker arm, assembling the arm in the engine, and obtaining records by means of a cathode ray oscilloscope. One of these records is shown in Fig. 33.

The picture is printed in such a manner that the high peak on the right-hand side corresponds to the rising flank of the cam, the one on the left, to the falling flank. The region between the two peaks represents spring load and



■ Fig. 33 – Cathode ray oscillogram of a valve rocker arm in service

forces resulting from a false valve motion. The interval shown is approximately 170 deg. By calibration, actual load values were determined.

In comparing electric strain gages with other methods of experimental stress analysis, a specific reference will be made to wire gages because of their widespread use.

The principal advantages of wire strain gages can be listed as:

(a) The gage may be left on the part indefinitely for checks at a later date.

(b) The gage is ideal for dynamic work, with a frequency range at least as high as 10,000 cps.

(c) Gage marks are not required on the test piece, making the gage well suited to polished aircraft parts.

(d) Since very little clearance is needed above the gage, it may be placed in otherwise inaccessible regions.

(e) The gage can be used at elevated temperatures up to 400 F.

(f) The strain readings may be made at a distance from the gage.

The principal disadvantages are:

(a) Waiting period between the time when the gage is applied and the time when strain readings can be taken.

(b) The possibility of an imperfect bond between the gage and the test piece.

(c) Lack of a specific calibration for each strain gage.

(d) Exploration for the direction of principal axes is not possible, unless rather cumbersome delta or strain rosette gages are used.

(e) The minimum gage length is limited to approximately 1/8 in.

II – TYPES OF PROBLEMS SUBJECT TO STRESS ANALYSIS

The three principal types of engine problems which can be studied by the methods of stress analysis are:

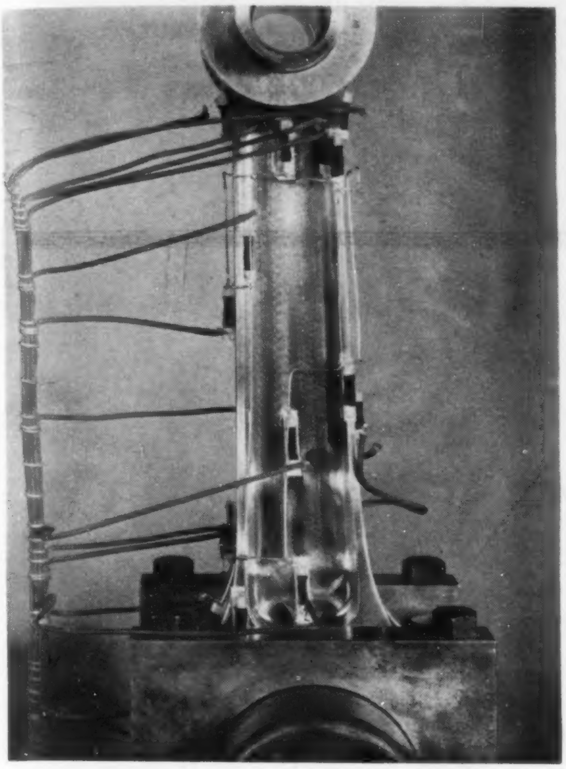
1. Evolution of a new design.
2. Determination of the cause of failure.
3. Comparison of various designs.

■ Evolution of a New Design

Experimental stress analysis should serve as a useful tool at the very inception of the problem. If the design is such that it can be subjected to a two-dimensional analysis, the work can begin the instant the drawing is completed. If the member is of a more complicated nature, stress analysis must wait construction of the actual part. The latter need not be finely polished and often not even finish-machined. Occasionally, though the design may specify a forging construction, the same part made as a casting can serve as a satisfactory model. The casting can be made of steel, aluminum, or even plastic, though the latter process is still in the experimental stage.

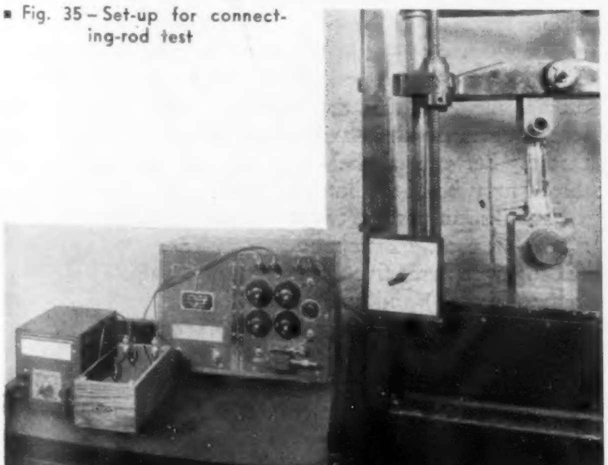
Stress analysis of a connecting rod provides an example of the study of a new design. Anticipated maximum service loads were calculated, both for tension and compression, from the reciprocating weights, engine speed, and gas pressure. These were used as reference loads; loads actually applied in the test were somewhat lower. The problem involved study of both the fork rod and the blade rod.

First, Olsen machine tests were conducted with Stresscoat in order to determine the location of highly stressed areas and stress directions. This was done for both tensile and compressive loads applied. Twenty-five electrical strain gages were then placed on each rod (Figs. 34 and 35).



■ Fig. 34 – Connecting rod mounted in an Olsen machine with electric strain gages in place

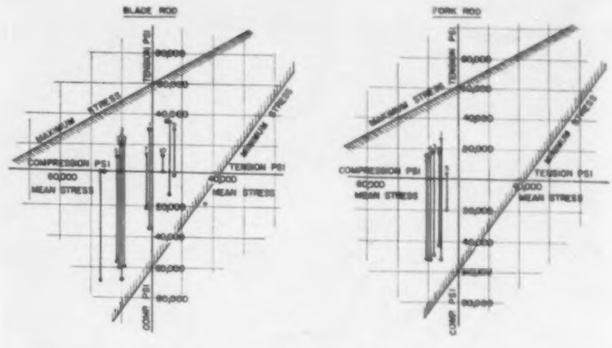
■ Fig. 35 – Set-up for connecting-rod test



Points of highest stress selected from the stress data thus obtained were then plotted on a Goodman diagram (Fig. 36).

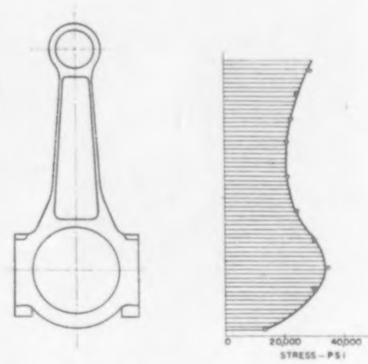
The graph shows that both the blade and the fork rod stresses are within the safety limit, although the stress distribution in the former is not as uniform as in the latter.

Fig. 37 shows an interesting point brought out in the present study. The graph represents a plot of tensile stress



■ Fig. 36 – Goodman stress diagram for the connecting rod

■ Fig. 37 – Stress analysis of the connecting rod



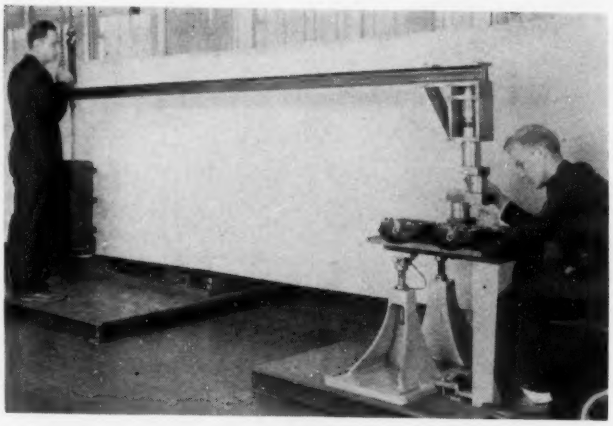
at the edge of the blade rod and shows the highest stress to be where the surface sweeps toward the bolt head. This is due to the flexibility of the section, and it results in bearing distortion under load. Although the tensile stress in this region is high, the corresponding compressive stress is low, so that the resulting stress range (point 8 in., Fig. 36) is not any higher than in the remaining connecting-rod sections.

■ Determination of the Cause of Failure

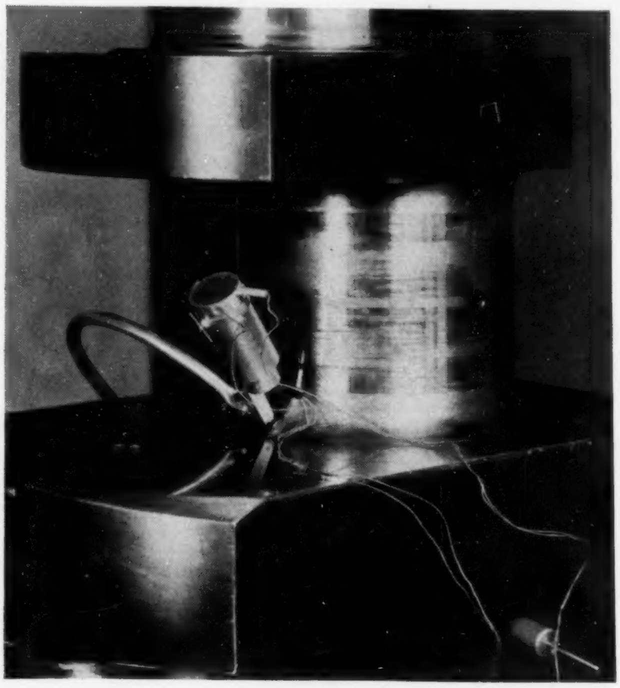
The method of experimental stress analysis can also be used to determine the cause of failure. This is illustrated by the study described below:

This investigation was undertaken in order to determine the cause of failure of two identical single-cylinder crankshafts, both failures having occurred in fillets. Inspection of the surfaces of the failed section pointed to a fatigue failure resulting from bending loads and originating at the rod journal fillet at the main axis of the cheek. The fracture progressed directly across the cheek into the fillet of the opposing journal.

A stress analysis was undertaken in order to determine whether these failures were due to features of design (metallurgically, the crankshaft was found to be satisfactory). The investigation was conducted on one portion of the broken crankshaft removed from the single-cylinder engine. To reproduce the bending system on the tested portion of the crank, the cheek in which failure occurred was welded to a mounting plate and the whole set-up assembled for loading as shown in Fig. 38. The measuring



■ Fig. 38 – Set-up for the stress analysis of a crankshaft



■ Fig. 39 – Photoelectric extensometer in place on the crankshaft

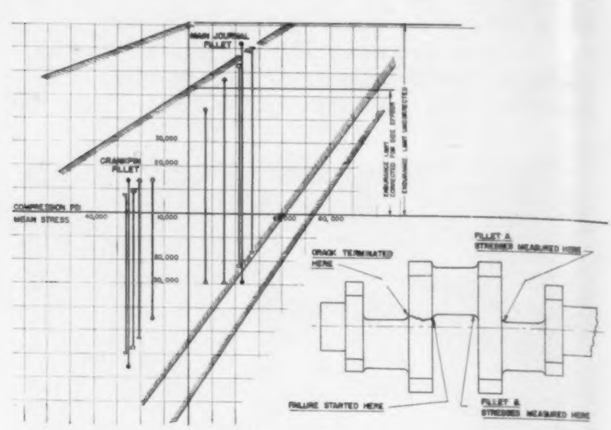
instrument was the photoelectric extensometer of 1/16-in. gage length, shown in place in Fig. 39.

Careful stress traverses were made in the crankpin fillet and the main journal fillets, for both the tensile and compressive loads applied. These data were then referred to the operating conditions by multiplying each stress value by the ratio of the operating (reference) load to the load actually applied on the test fixture.

The resultant stress values were then plotted in a Goodman diagram for several operating conditions (Fig. 40).

In constructing this diagram it was found necessary to consider the size effect. On the basis of data available in literature, it was found necessary to decrease the endurance limit (as determined on a Moore machine) by about 30%.

The experimentally determined stress range for both the crankpin fillet and the main journal fillet, when thus plotted, shows that while the former (corresponding to the fillet which did not fail) is well within the allowable stress range, the latter (failed fillet) exceeds its safety limit. Therefore, within the error involved in our knowledge of



■ Fig. 40 – Goodman stress diagram for the crankshaft fillets

the actual endurance limit, the crankshaft failure can be attributed to a defective design. The prevailing engine loads appear to be excessive for the design used.

■ Comparison of Various Designs

In many cases a number of alternative design proposals are submitted, and the problem is to select the cheapest and best design. It is in this field of comparison that the experimental method of stress analysis is most effective.

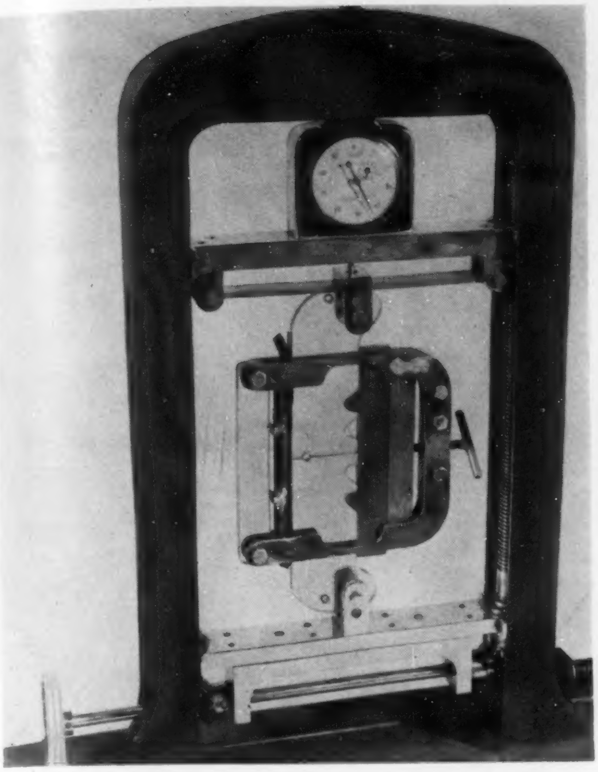
The other two applications of an experimental stress analysis (see "Evolution of a New Design" and "Determination of the Cause of Failure") suffer from the disadvantage that unless the service loads involved and the characteristics of the material used are known, absolute stress values cannot be obtained. Problems of comparison between alternative designs do not demand this information, and therefore their solution can be applied more readily to actual practice. One of such problems is described below:

The purpose of this investigation was to compare various thread forms for an application involving essentially two tubular members threaded together. The loads produced by this device result in axial and thrust forces acting on the thread. Because of the latter, it was specified that the thread form be of the buttress variety.

For this purpose a two-dimensional photoelastic technique was used. Plastic models representing various thread designs were made five times size, then placed in a loading frame (Fig. 41) and analyzed in a polariscope. The first few tests were conducted with a six-thread train. This, however, was soon abandoned in favor of a single thread, so that various designs could be studied on a comparative basis. For this purpose, all the threads investigated were reduced to a common six-pitch scale.

In all the specimens tested, the axial load was constant (65 lb). The magnitude of the applied thrust, however, depended on the thread design, since some threads demanded higher thrust to hold them together than others. Thus, six of the nine designs studied had a 5-lb thrust, while the remaining three had 30 lb. This load was selected as just sufficient to hold the engaging threads together.

In the actual assembly, the two threads are made of different metals. To simulate this condition, one plastic specimen was made of opaque bakelite, the other of trans-



■ Fig. 41 – Plastic thread model in the loading frame

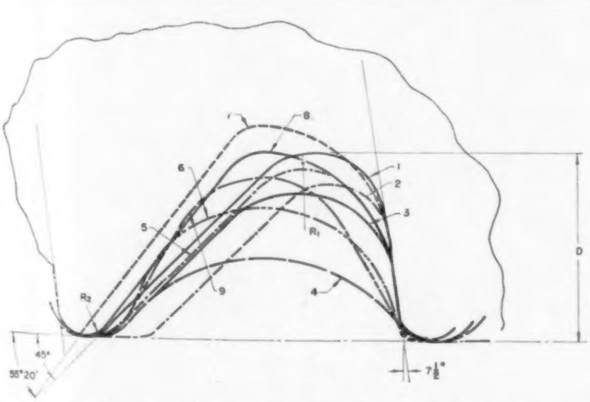
parent catalin, the ratio of the elastic moduli of the two being the same as for actual parts.

Reasonable precautions were taken to eliminate the effect of other variables. Thus the effect of thread friction was minimized by an insertion of a thin film of oil between the engaging plastic threads. To decrease the inaccuracy of the thread form, plastic specimens were prepared from machined metal templates, made to specified dimensions, and lapped carefully together.

In this manner nine different thread forms were studied. The form contour of each thread and the resultant stress data are given in Figs. 42 and 43 respectively.

It will be noted that two stress values are given for each thread:

(a) Contact stress
(b) Fillet stress



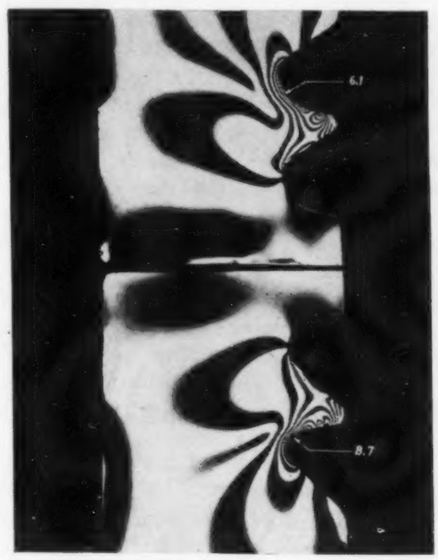
■ Fig. 42 – The nine thread forms studied – all threads reduced to 6 pitch

THREAD DESIGN					STRESS				LOAD APPLIED	
					Values — p.s.i.		% improvement			
No.	Description	R_1 (in.)	R_2 (in.)	D (in.)	Contact	Fillet	Contact	Fillet	Axial (lbs.)	Thrust (lbs.)
1	DESIGN No. 1	.034	.016	.903	1950	1520	0	0	65	5
2	DESIGN No. 2	.028	.013	.075	1870	1530	4.1	-0.7	65	5
3	DESIGN No. 3	.046	.025	.0714	1530	1350	21.5	11.2	65	5
4	DESIGN No. 4	.075	.025	.040	2120	1180	-8.7	22.4	65	30
5	DESIGN No. 5	.029 .063	.019	.082	1690	1510	13.3	0.1	65	5
6	DESIGN No. 6	.052	.018	.052	1780	1330	8.7	12.5	65	30
7	DESIGN No. 7	.0173 .0617	.007	.103	1350	1170	30.8	23.0	65	5
8	DESIGN No. 8	.0173 .0517	.014	.0900	1350	1170	30.8	23.0	65	5
9	DESIGN No. 9	.0443	.022	.0746	2370	1190	-21.5	21.7	65	30

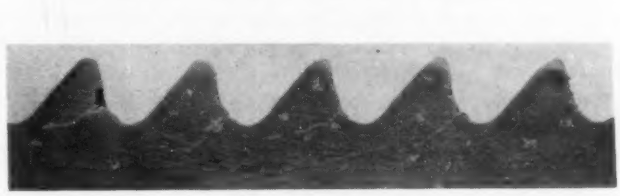
■ Fig. 43 – Stress analysis of the nine thread forms

Contact stress represents maximum stress resulting from the face-to-face contact of the engaging threads. Fillet stress represents maximum stress resulting from the deflection of the loaded tooth, and is located in the region void of actual thread contact. Neither value determines principal stresses since the existing loading conditions eliminate free boundary even at the filleted section.

In interpreting the experimental data, both stress values must be taken into consideration since neither one by itself can be taken as an index of the thread strength. In every case studied the contact stress was the higher of the two and its location corresponded quite well to the region of actual failure. (Compare Figs. 44 and 45.) However, because of insufficient background of the relationship between stress and fatigue failures, it was decided to interpret



■ Fig. 44 – Photoelastic view of thread model



■ Fig. 45 – Model of thread forms showing regions of actual failure

stress data in terms of both the contact stress and the fillet stress.

Of the nine thread forms studied, thread designs Nos. 1, 3, and 7 merit closer attention. Using design No. 1 as reference, stress data show the following percentage improvements for the remaining two threads:

	On the Basis of	
	Contact Stress	Fillet Stress
	%	%
Design No. 1	0	0
Design No. 3	21.5	11.2
Design No. 7	30.8	23.0

On the basis of the present investigation, design No. 7 appears to be more satisfactory than any other thread form studied. This, of course, does not imply that this design will necessarily be best in actual operation, because many operating factors could not be duplicated in the present study. Once the correlation between stress and actual performance is known, a clearer insight as to the relative merit of each thread will be obtained.

Following this investigation, design No. 3 thread was incorporated in actual assembly and tested in service. Reports received so far indicate that this thread lasts longer than the original one (design No. 1).

III — STRESS ANALYSIS OF A TYPICAL PART: CRANKSHAFT

Due to the complicated nature of loading, crankshaft design represents one of the most difficult problems in the field of mechanics. The design involves consideration of lateral and torsional vibrations, of crankcase rigidity, and of stress concentration. As a result, an exact analytical solution is impossible and frequently recourse must be taken to supplementary methods of experimental measurements.

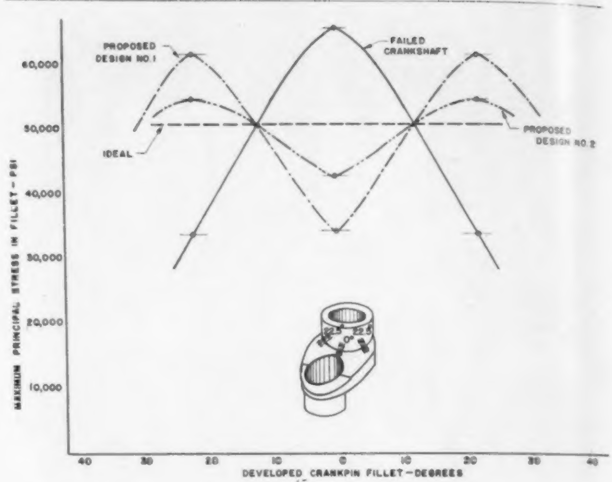
This is particularly true in the consideration of crankcase stresses, since the majority of crankshaft failures are due to increased stress conditions in fillets or oil holes. It is in this particular application that the experimenatl means of stress analysis is found to be most effective.

In the following sections, some of the stress problems conducted in Chrysler laboratories in connection with crankshaft development are discussed in detail.

■ Solid Versus Hollow Main Journal

This work was undertaken as a result of crankshaft failure in a twin-cylinder engine. The failed shaft had a solid main journal and a hollow cylindrical pin, failure originating at the crankpin fillet in the plane of the throw. The shaft designed to replace it (proposed design No. 1, Fig. 46) had hollow cylindrical holes both in the main and in the crankpin journals. The object of this investigation was to determine whether the new shaft would be an improvement.

For this purpose an experimental stress analysis was conducted on single-throw crankshafts. The experiment consisted of supporting each shaft at its main journals and loading it at the top dead-center position at the pin through the connecting rod. All the stress data were taken with the photoelectric extensometer.



■ Fig. 46 – Stress distribution in crankshaft fillets – comparison between hollow and solid journals

For the reference load used, the following stresses were obtained:

Crankshaft Design		Stresses, psi Plane of Throw	$22\frac{1}{2}$-deg Plane
Failed Crankshaft	Solid Main, Hollow Pin	66,000	34,200
Proposed Design No. 1	Hollow Main, Hollow Pin	33,400	62,000

The failed crankshaft ran in the engine for 12,000,000 cycles. This is, therefore, a case of fatigue failure (as evidenced, in addition, by the appearance of the failed section), the measured stress (66,000 psi) representing approximately the endurance limit of the crankshaft.

Data given in the above table show that hollowing the main journal has decreased maximum stress from 66,000 psi to 62,000, thus producing a 6% improvement. This is rather a small change and its meaning can probably be clarified by the following consideration:

Stress studies (described later in section "Determination of Critical Stress Regions") indicate that when a shaft is subjected to bending loads, stress distribution along the crankpin fillet traverse depends upon the lightening hole size. As the hole size is increased, the point of maximum stress begins to be shifted away from the center (0 deg); the stress at the center becomes lower, the stress farther away ($22\frac{1}{2}$ deg), higher.

This is confirmed by the present study (Fig. 46): the failed crankshaft (solid main) is characterized by the maximum stress (66,000 psi) being in the center (0 deg). In the proposed shaft (proposed design No. 1), maximum stress has shifted to a $22\frac{1}{2}$-deg plane, leaving the stress at the center considerably lower.

The two shafts apparently represent two extreme cases, and consequently a more favorable stress distribution should be realized from an intermediate set of dimensions. The shaft designed according to this principle (proposed design No. 2), when stress analyzed, did show a definite improvement (20%). For the "ideal" shaft the two dimensions should be decreased probably still further, but in view of insufficient background in fatigue data it has been decided to recommend the dimensions given above instead.

Standard Versus Undercut Fillets

The purpose of this investigation was to compare undercut fillets (of the type incorporated in the Junkers-Jumo crankshaft) with standard circular fillets. In the first phase of this work, a full-size two-dimensional plastic model was loaded as shown in Fig. 47 and analyzed in a polariscope. From the resultant stress photographs, data were computed and plotted as shown in Fig. 48. An identical test was conducted with undercut crankpin fillets incorporated in the model. The resultant data have shown the maximum fillet stress to be the same in both cases.

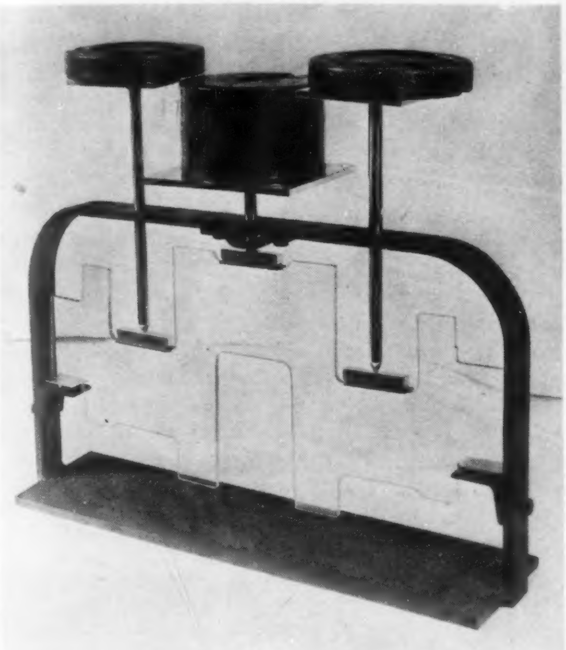


Fig. 47 – Two-dimensional plastic model of crankshaft in the loading frame

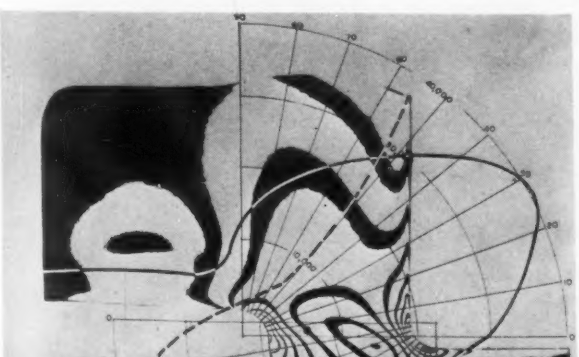


Fig. 48 – Stress analysis of crankshaft fillets

The second phase of this investigation involved loading an actual crankshaft in bending, and measuring fillet stresses by means of the photoelectric extensometer. The procedure and method of testing were similar to those described in section "Determination of the Cause of Failure." The purpose of the test was to check the results previously obtained by the photoelastic method.

The resultant data given in Fig. 49 show that both the standard and the undercut fillets give the same maximum stress and thus the use of the Junkers-Jumo fillet would not provide an improvement in the crankshaft under consideration.

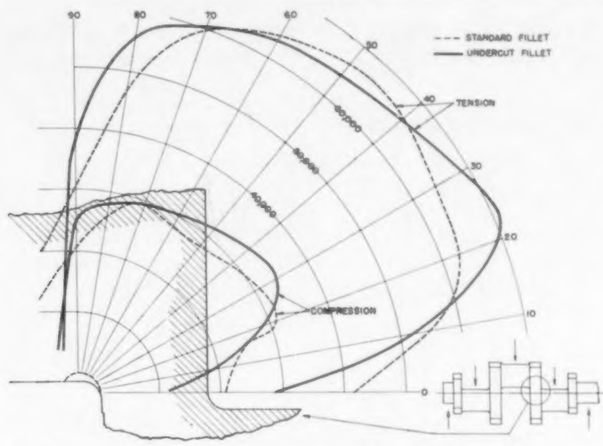


Fig. 49 – Stress distribution in crankshaft fillets – comparison between standard and undercut fillets

All that the stress data obtained in the current test imply is that an undercut compound fillet does not offer any advantage over a standard circular fillet for the crankshaft under consideration. Lack of sufficient stress or fatigue background precludes the possibility of generalizing this statement for different crankshafts.

An apparent discrepancy will be noted between absolute stress values obtained photoelastically and those obtained by means of extensometers placed on the actual crankshaft. Also, the form of the stress distribution curves does not appear exactly the same in both cases. Although this does not affect the validity of the current investigation in which only a comparison test was in question, the discrepancy demands some explanation:

The photoelastic test was conducted on a two-dimensional replica of the actual crankshaft. The extensometer measurements were taken on the shaft itself. The two-dimensional analysis gives stress in the fillet as if the fillet extended straight across the cheek; thus, the stress obtained for the fillet was an average stress distributed over the 6-in. cheek width. Contrarily, the three-dimensional investigation determined peak stress in the fillet in the center of the cheek, the stress dropping on each side of the centerline and gradually diminishing to the edges of the cheek. Therefore the peak stress obtained by the extensometer yielded a larger stress value.

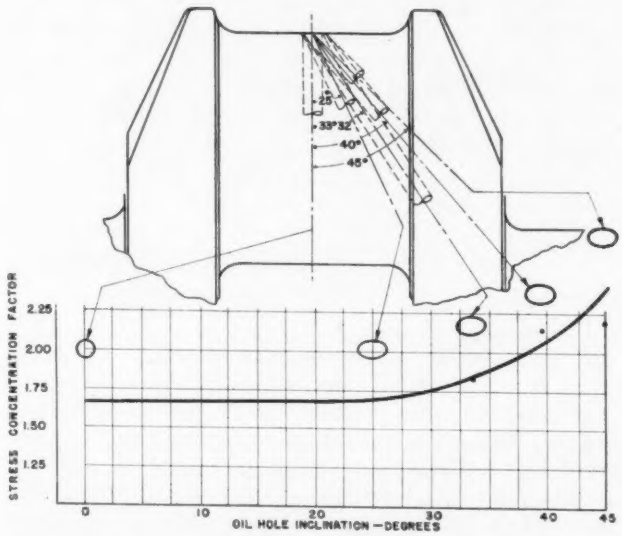
Effect of Oil-Hole Inclination

The purpose of this investigation was to determine the effect of the oil-hole angle on the stresses in the crankpin. For this purpose a two-dimensional photoelastic method

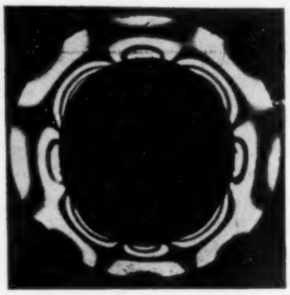
was used. The test was performed on a plastic model representing a plan view of the intersection of the oil hole with the crankpin. In the case of a radial hole, the intersection results in a circle; for a diagonal hole, the intersection produces an ellipse.

The test was conducted for various oil-hole inclinations, in each case the exact hole dimensions being obtained by a careful projection. Plastic models were then loaded in shear in order to simulate conditions existing around the oil hole of the actual crankpin. For each hole two different loads were used in order to insure higher accuracy.

The resultant data (Figs. 50, 51, and 52) show that for a given crankshaft design no difference in crankpin stresses exists for the range of oil-hole angles from 0 to 25 deg. From 25 to 45 deg, stresses increase, so that a 45-deg oil hole produces about 30% higher stresses than a radial hole.



■ Fig. 50 – Stress analysis of crankpin oil hole – for various oil-hole inclinations



■ Fig. 51 – (left) Photoelastic view of 25-deg inclined hole – average stress = 172 psi (major axis = 1.203, minor axis = 1.906)

■ Fig. 52 – (right) Photoelastic view of 25-deg inclined hole – enlarged

It is clearly recognized that these data do not represent actual conditions existing in the engine because of the two-dimensional nature of the test. It is felt, however, that they are sufficiently indicative of these conditions to warrant the general conclusion that from the stress viewpoint no advantage can be realized by substituting a diagonal oil hole for a radial one.

■ Determination of Critical Stress Regions

The purpose of this investigation was to locate and determine critical stress regions in a crankshaft. A similar study had been previously reported by the General Motors Research Laboratories Division.

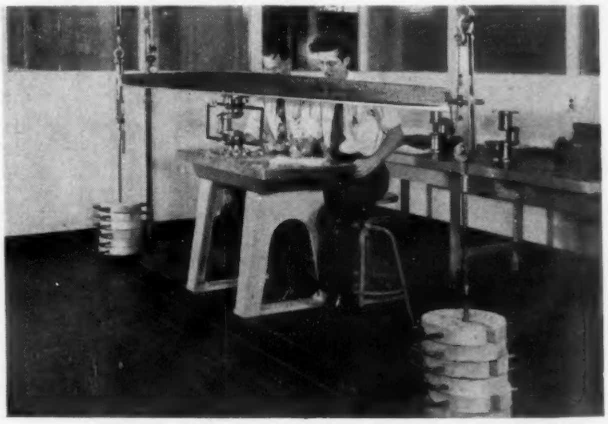
For this test a single-throw section of a crankshaft was used, since a whole crankshaft was not available. This step appears to be justified since the investigation was primarily concerned with the interrelationship among oil holes, fillets, cheeks, and so on, rather than the effect of other variables, such as the effect of one throw on the stresses of another. In order to accomplish the latter objective, stress measurements should be made on a full crankshaft mounted in a crankcase unless the support offered by the case could be successfully simulated. Since such tests often entail considerable difficulty (for instance, mounting the extensometer on a crankshaft assembled in a crankcase) they should be preceded, whenever possible, by stress studies on single-throw crankshafts. Such tests are useful and necessary because:

(a) They comprise the first step toward a broad crankshaft program.

(b) They provide knowledge of the trend of the crankshaft strength as a function of its design.

(c) They yield data on stress distribution in a complete crankshaft as a function of a limited number of variables.

The loading fixture used in the current investigation is shown in Fig. 53 (the machine being set up, at the time, for the measurement of crankshaft rigidity). The fixture was constructed in such a manner that either bending or torsional load could be easily applied.



■ Fig. 53 – Loading fixture set up for measuring crankshaft rigidity

The investigation involved Stresscoating the crankshaft, and then placing the 1/16-in. photoelectric extensometer at various critical regions located by the Stresscoat cracks. At each point strains at right angles were measured, from which stresses were calculated by the conventional stress formulas:

$$S_x = \frac{E}{1 - \mu^2} (\epsilon_x + \mu \epsilon_y)$$

$$S_y = \frac{E}{1 - \mu^2} (\epsilon_y + \mu \epsilon_x)$$

In this manner stresses due to torsional and bending loads were separately measured. Stress values were then

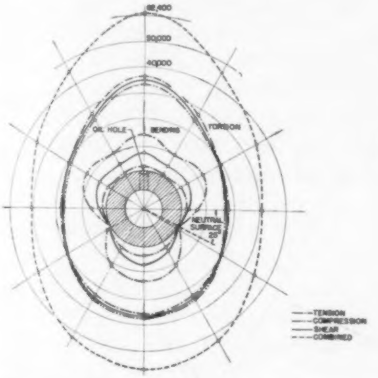
combined according to the Von Mises-Hencky theory in order to obtain the equivalent or combined stresses:

$$S_e = \sqrt{S_x^2 + S_y^2 - S_x S_y + 3 V^2}$$

Where S_e = equivalent or combined stress at a point
S_x, S_y = normal stresses due to bending loads
V = shear stress due to torsion

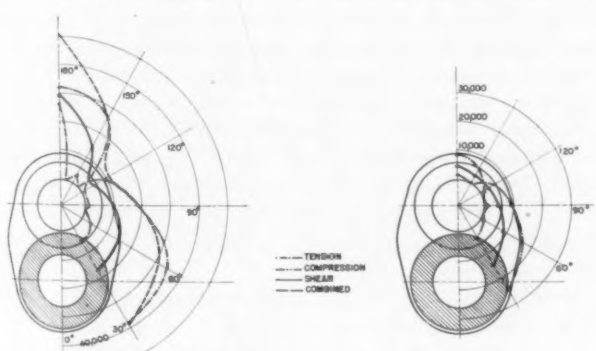
Stresses were thus determined for various critical stress regions of a single-throw crankshaft, the resultant data being then plotted in the manner shown in Figs. 54 to 58. A review of these graphs indicates the following:

(a) *Crankpin (Fig. 54)* – Maximum equivalent stress occurs at the oil hole, the trajectory of equivalent stresses being similar to that due to torsion. The effect of the bending load is negligible as attested by the observed value of the ratio between the equivalent and the torsional stress ($\sqrt{3}$). In torsion, tensile and compressive stresses are equal, thereby producing pure shear.



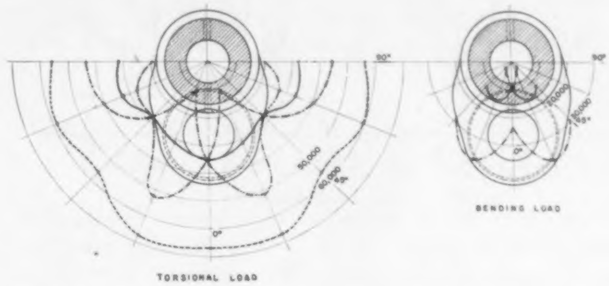
■ Fig. 54 – Stress analysis of crankshaft – crankpin

(b) *Crankpin Lightening Hole (Fig. 55)* – These measurements were taken as near the edge of the lightening hole as possible. The trajectory of equivalent stresses again resembles that of torsion. Minimum stress will be observed at 120 deg, thus indicating that in this region only some of the material is structurally necessary.



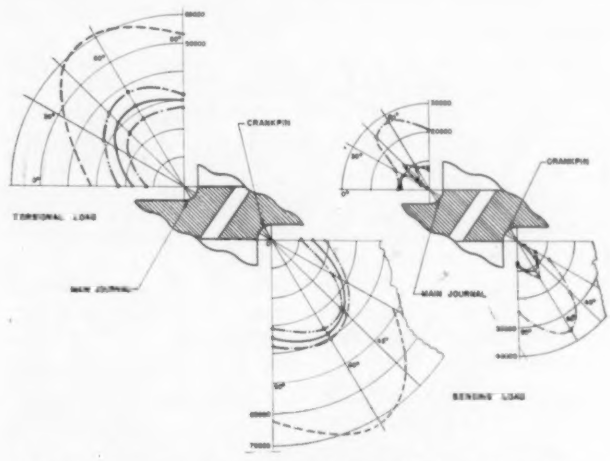
■ Fig. 55 – Stress analysis of crankshaft – traverse of crankpin lightening hole

(c) *Traverse of Crankpin Fillet (Fig. 56)* – Both in torsion and in bending, maximum tensile stress occurs in the 22½-deg plane rather than in the plane of the crankpin throw (0 deg). In the plane of the throw, tensile and compressive stresses due to torsion are equal, thus resulting in pure shear. Maximum equivalent stress occurs in the 0 to 22½-deg interval, and all the observed fillet failures took place in this region.



■ Fig. 56 – Stress analysis of crankshaft – traverse of crankpin fillet

(d) *Crankpin and Main Journal Fillets (Fig. 57)* – In all the cases studied stresses in the crankpin fillet were found to be equal or greater than those in the main journal fillet. Tensile and compressive stresses due to torsion are equal, thus producing pure shear, as expected from theoretical considerations.

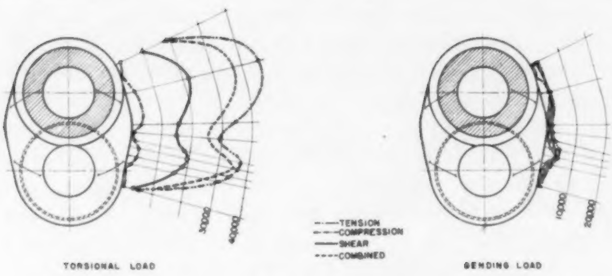


■ Fig. 57 – Stress analysis of crankshaft – fillets in the plane of the throw

(e) *Traverse of Cheeks (Fig. 58)* – Two peaks were found to exist, with a low region in between. The small stress observed in the cheek section, approximately midway between the crankpin and the main journal, indicates that, contrary to expectation, in the present design this section need not be any wider than the remaining regions. Better stress distribution could be realized by removing metal in the low stress region.

■ Various Lightening-Hole Designs

The purpose of the investigation described in the previous section was to obtain a stress pattern in terms of which various crankshaft designs could be compared. The specific need for such information arose out of a consideration of various lightening-hole designs associated with a particular engineering development.



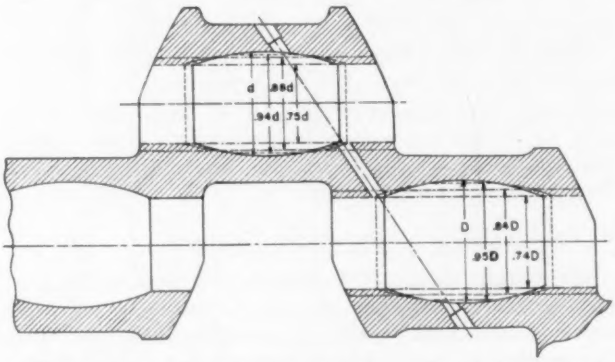
■ Fig. 58 – Stress analysis of crankshaft – traverse of cheeks

Standard American practice in airplane crankshaft design involves the use of cylindrical lightening holes both in the crankpin and in the main journal. A series of fatigue tests conducted in Europe indicate, however, that a higher crankshaft strength may be realized by incorporating barrel-shaped lightening holes. From a manufacturing point of view, barrel-shaped holes result in considerably higher cost, and the advantages of such designs must be real and substantial to be economically justified. Stress data are not available concerning the comparison between cylindrical and barrel-shaped lightening holes. The current investigation was therefore undertaken for two reasons:

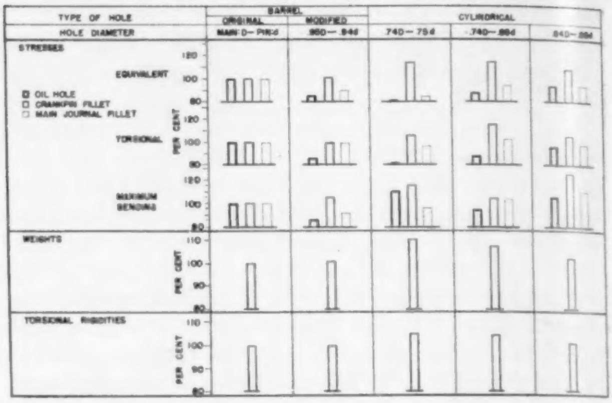
1. To compare barrel and cylindrical lightening holes.
2. If the barrel-shaped hole does not offer any real advantage, to develop a cylindrical hole which would give the highest crankshaft strength per unit of weight.

Current investigation involved a repetition of the stress measurements described in the previous section for each crankshaft design studied. Not all the stress regions or stress trajectories were repeated because it was soon verified that the design comparison could be safely limited to the study of the oil-hole and fillet stresses. The five designs studied are shown in Fig. 59 and the principal data derived in the present series of tests in Fig. 60.

A review of these data indicates that the use of barrel-shaped lightening holes does result in lower crankshaft stresses. For the same unit weight, however, and the same rigidity, less than 10% reduction in stresses was realized (original barrel versus cylindrical (0.84D — 0.88d), Figs. 59 and 60). Whether this improvement justifies the increased cost involved in the adoption of barrel-shaped holes depends on personal judgment and the circumstances involved.



■ Fig. 59 – Stress analysis of crankshaft – various lightening hole designs



■ Fig. 60 – Stress analysis of crankshaft – comparison between lightening hole designs

IV — CONCLUSION

Development of experimental stress analysis has now reached the point where it has become a practical tool in the hands of an engineer or a designer. Four methods are available, and in the utilization of these methods there are two principal means of procedure: static tests, using simulated service loading, and dynamic tests under actual service conditions.

Static load testing is usually the most convenient and generally favored because the problem of instrumentation is greatly simplified. The penalty for this simplification lies in the fact that the investigation is often conducted on the basis of loads which may or may not correspond to operating conditions. Dynamic load testing demands more complex instrumentation, and so far it is less thoroughly developed. A comprehensive stress analysis investigation requires both phases of testing: dynamic tests should establish the mode and magnitude of operating loads, while static measurements will determine the corresponding stresses. Refinements of instrumentation and technique are necessary to promote greater accuracy and speed, but this particular phase of experimental stress analysis is developing in a satisfactory manner.

Complete solution of the problem of design demands, in addition, knowledge of the properties of the material used. In this field, however, we know the least, and many aspects of this science have yet to be studied and classified. Essential information is still lacking for the following problems:

1. The influence of size when translating fatigue data from small specimens to larger parts.
2. Evaluation of the effect of combined stresses in terms of the fatigue data derived from rotating-beam testing.
3. The influence of preload and operating stress range on the endurance life.
4. The effect of stress-raisers on the mechanical properties of materials.
5. The effect of manufacturing processes on the endurance life.
6. The mechanical properties at elevated temperatures.

As this paper has attempted to show, stress analysis, even in its as yet partially developed stage, can provide essential information involving strength of materials. As the technical and fundamental concepts of this science are further developed, the goal of maximum strength at minimum cost and weight will be coming closer to realization.

PISTON-RING SCUFFING as a Criterion of OIL PERFORMANCE

by **GEORGE H. KELLER**
Wright Aeronautical Corp.

THE high-output aircraft engine, as we know it today, represents the result of the combined researches of the metallurgist, the petroleum chemist, and the engine designer. Their work, over a 10-year period, has resulted in an engine with a 100% increase in power output, a 50% reduction in specific weight, and a 20% reduction in specific fuel consumption. The petroleum chemist's more recent contributions have been fuels of increased octane rating, and oils having improved lubricating properties and better stability. The improvements in fuels and the resulting possibilities for engine improvement have been quickly recognized and utilized, the result being higher permissible maximum power outputs and lower specific fuel consumptions for long-range cruising operation. The improvements in lubricating oils have not been as spectacular as those in fuels, but nevertheless there is definite evidence that better aircraft-engine lubrication (mechanical condition and cleanliness) has resulted from the research on aircraft-engine oil. The improved lubricants have been tested, proved, and adopted by commercial airline operators during the past seven years, but up to the present time no move has been made to ensure the supply of lubricating oil for military aircraft that incorporates the proved developments in oils.

CONDITION of engine parts after testing under standardized procedure is the only satisfactory evidence for comparing the lubrication qualities of different oils, Mr. Keller concludes, showing the detailed set-up for such engine testing.

Of seven pertinent operating variables, all except cooling conditions and oil supply were held constant. The five were power, speed, fuel mixture strength, detonation, and cylinder, piston, and ring design.

Because experience has shown that detonation can destroy satisfactory cylinder lubrication and lead to excessive wear and ultimate failure, fuel having an octane rating of iso-octane + 0.8 cc tel (Aviation Test Method) was used for most of the test work reported. A Waukesha Cooperative Universal Engine fitted with a Wright Cyclone C9GC production cylinder and piston assembly, 6 1/8 x 6 7/8 or 202 1/2 cu in., was used.

Prior to the test the cylinder barrel was lapped with Minnesota 3M compound No. 500-2-A, with an old piston and old rings as the lapping surface. This was to eliminate any surface deposits from the previous test. The oil system and interior of the engine were flushed with solvent naphtha and then with the oil to be tested. A fresh charge of oil was used for the run-in and for the test itself.

Combined researches of metallurgists, petroleum chemists, and engine designers during the past 10 years, the author says, has resulted in an engine with a 100% increase in power output, a 50% reduction in specific weight, and a 20% reduction in specific fuel consumption.

THE AUTHOR: GEORGE H. KELLER (J '37), assistant project engineer of the Wright Aeronautical Corp., received a B.S. degree in Mechanical Engineering from Lehigh University in 1933, and an M.S. degree in Mechanical Engineering from Pennsylvania State College in 1934. From 1934 to 1939 he was research assistant at Pennsylvania State College, working on a lubricating oil development project sponsored jointly by the Pennsylvania Grade Crude Oil Association and the college. This work covered tests in a wide variety of automotive and single-cylinder laboratory test engines. Mr. Keller joined Wright Aeronautical Corp. in 1940 as a test engineer, and later that year was made assistant project engineer of the Fuels and Lubricant Unit, which involves the testing of fuels and lubricating oils to determine their suitability for use in aircraft engines.

In order to learn how some of the improvements in aircraft-engine lubricating oil could be used to advantage in engines operated under borderline conditions, and to show that under such conditions oils meeting current specifications do have appreciably different lubricating properties, considerable work has been done with a full-scale, single-cylinder, aircooled aircraft engine in which most of

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

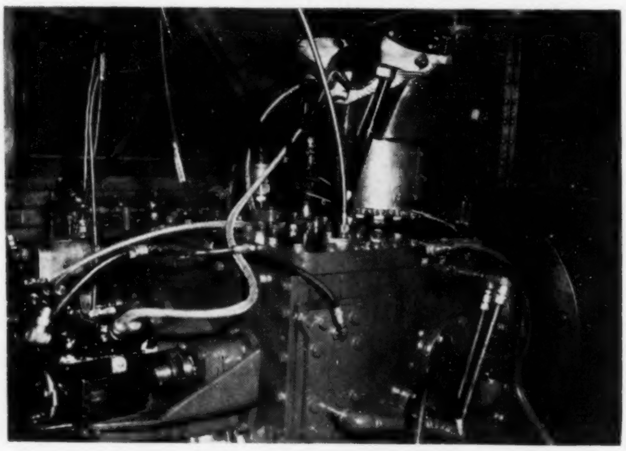
the important variables could be evaluated and controlled. An analysis of the problem of cylinder lubrication, based on test and experience, indicates that the following operating variables are pertinent:

1. Power
2. Speed
3. Fuel mixture strength
4. Cooling conditions
5. Oil supply to the piston and cylinder
6. Detonation
7. Cylinder, piston, and ring design

■ Cooling and Oil Supply Varied

For test purposes, it was decided to hold all of these variables constant except the cooling conditions and oil supply, arriving at test conditions where there was an appreciable difference between the performance of oils. This was done by successively increasing the severity of operating conditions, which in turn was accomplished by decreasing the amount of cooling and the oil supply to the piston and cylinder wall. The power, speed, and mixture were chosen to correspond with those in actual use for present day maximum power (take-off) conditions. Since experience shows definitely that detonation can destroy satisfactory cylinder lubrication and lead to excessive wear and ultimate failure, a fuel having an octane rating of iso-octane + 0.8 cc TEL by the Aviation Test Method was used for most of the work. Another precaution taken to rule out any effects of detonation was to limit the cylinder-head temperatures, as measured by a thermocouple in the rear spark plug gasket, to 400-425 F. The cylinder, piston, and rings used were current production parts, chosen because of availability and their being exemplary of current high-output designs.

The engine used for this work was a Waukesha Cooperative Universal Engine fitted with a Wright Cyclone C9GC production cylinder and piston assembly. A view of the installation is shown in Fig. 1. The cylinder has



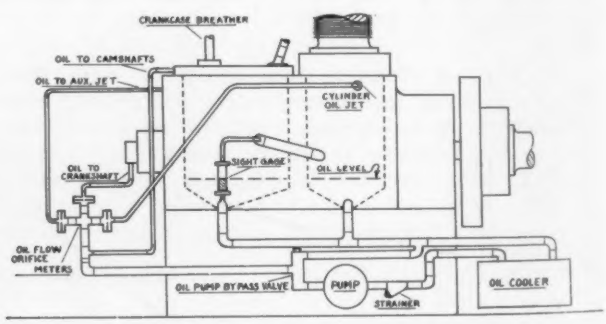
■ Fig. 1 – Waukesha engine showing baffles and barrel insulation

a bore of $6\frac{1}{8}$ in. and a stroke of $6\frac{7}{8}$ in., resulting in a piston displacement of $202\frac{1}{2}$ cu in. The piston was fitted with six rings, all having tapered faces. Five rings were located above the piston pin and one below, near the bottom of the skirt. The top three rings were of the wedge type and the lower three rings had parallel side faces. The rings were installed so that the tapered faces of the top five rings scraped down and the bottom ring scraped up. The fourth and fifth ring grooves were provided with oil drain holes. This piston and ring design has been commonly referred to as the "uniflow type."

Fig. 1 shows the cooling air baffles, so arranged as to direct the cooling over the cylinder head only. This arrangement was adopted after early tests showed that, in order to get the high cylinder-barrel temperatures desired, no cooling blast was needed on the barrel. For the actual test itself, the cylinder barrel was insulated with a polished aluminum jacket to reduce the heat loss by radiation and convection, thereby maintaining the temperatures desired for the test. In order to reduce the conduction of heat from the cylinder to the relatively cool mass of the engine, the cylinder adapter flange was fitted with two 1000-w electric heaters. The heaters acted as an effective heat dam, and made it possible to maintain a satisfactory temperature gradient from the top of the cylinder to its base. The aluminum insulating jacket can be seen in position on the cylinder in Fig. 1. The so-called "mid-barrel" temperatures were measured by four thermocouples, silver soldered directly to the cylinder barrel at the middle row of finning, and located at 90-deg intervals around the barrel. The cylinder-base temperatures were measured by four standard base thermocouples of the "peened-in" type located in the fillet just above the cylinder-base flange, directly below the "mid-barrel" thermocouples.

■ Type of Lubrication System Used

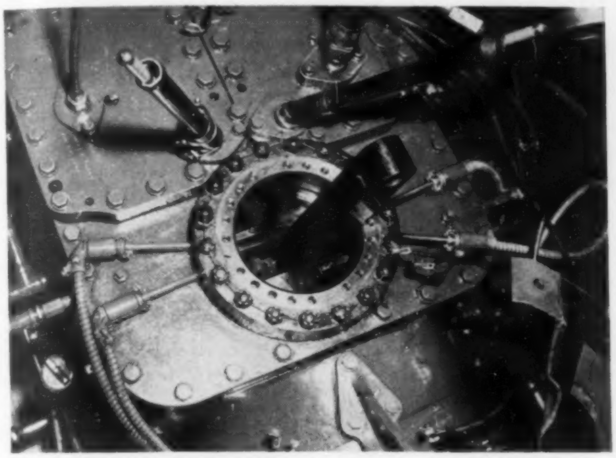
A sketch of the lubricating system used is shown in Fig. 2. The system was designed to operate with an oil



■ Fig. 2 – Sketch of minimum-capacity oil system

volume of 2 gal, this total quantity of oil in circulation being comparable to the volume per cylinder used in full-scale multicylinder engine installations. Cylinder lubrication in this engine was normally supplied by the throw-off from the crankpin and stationary jets located directly beneath the cylinder. During the course of the development of this test method, it was found necessary to install a baffle between the engine crankcase and the underside of the cylinder in order more accurately to control the oil flow reaching the cylinder walls and piston. This baffle was located just below the bottom of the cylinder and contained a slot just wide enough to clear the movement

of the connecting rod, thereby effectively isolating the cylinder lubrication from the oil throw-off of the crankpin. Fig. 3 shows the baffle with the cylinder and piston removed. The oil for lubricating the cylinder was supplied by two jets located between the baffle and the bottom of the cylinder skirt. The main oil supply jet, used for all running, was located on the thrust side of the cylinder. An auxiliary jet was located on the antithrust side. This jet



■ Fig. 3 – Oil baffle, cylinder, and piston removed

was used to supply additional oil during engine warm-up and break-in periods. Both of these jets discharged the oil as a solid stream directed to strike the underside of the piston. The cylinder walls were lubricated by splash from the underside of the piston. The oil flows to the connecting-rod bearing, and jets were measured by means of sharp-edge orifice meters and controlled by regulating the pressures. All parts of the engine, including the rocker boxes, were lubricated from the same oil supply which lubricated the cam and power sections. The oil pressure was supplied by an electrically driven pump, and the oil temperature was controlled by an external cooler. The engine crank and cam cases acted as a wet sump.

■ Preparation for Tests

Before each test, the cylinder barrel was lapped with Minnesota Mining and Manufacturing Co.'s 3M compound No. 500-2-A, a fine-grit emery compound in the form of an oil-mix paste, using an old piston and old rings as the lapping surface. The barrel was lapped to eliminate any surface deposits from the previous test in order that all tests would be started with the same character of surface finish on the cylinder walls.

The oil system and the interior of the engine were flushed before the beginning of each test with solvent naphtha and then with the oil to be tested. A fresh charge of oil was used for the run-in and for the test itself.

The run-in for each test was made under the following conditions:

Oil-In Temperature	180-190 F
Crankpin Oil Flow	6.5 lb per min at 2500 rpm
Head Temperature (rear spark plug gasket)	375-385 F
Cylinder Base Temperature	265-285 F
Oil Flow to Cylinder	7.5-8.5 lb per min

Run-In Schedule:

Min	Bmep	Bhp	Rpm	Specific Fuel Consumption
10	58	22	1500	0.70-0.75
10	102	44	1700	0.70-0.75
20	145	72	1950	0.70-0.75
20	172	95	2150	0.70-0.75
45	189	111	2300	0.70-0.75
45	200	122	2400	0.70-0.75
30	210	134	2500	0.70-0.75
300	140	72	2000	0.70-0.75

The 5 hr at cruising power were added to the usual 3-hr run-in in order that small differences in rings and the manner in which they seated themselves would be more nearly eliminated.

After the run-in, the oil system was drained and refilled with test oil. The cylinder barrel was wrapped with the aluminum radiation shield and the test proper was run under the following conditions, after a 30-min warm-up:

Time	5 hr
Rpm	2500
Bmep	210
Bhp	134
Specific Fuel Consumption	0.74-0.75
Head Temperature (rear spark plug gasket), F	400-410
Base Temperature, F	400-410
Oil-In Temperature, F	180-190
Oil Flow to Cylinder, lb per min	1.25-1.50
Crankpin Oil Flow, lb per min	5.5-6.5
Mid-barrel Temperature, F	480-490 (average)

■ Factors Studied

The relative merits of the oils under test were determined by the following factors:

1. Condition of rings and piston following the test.
2. Oil consumption during the test.
3. Blowby during the test.

It was found that, of the several oils tested, the better oils could complete the 5 hr of high-power running with negligible oil consumption while the poorer oils could not be run the entire period because high oil consumption dropped the oil level below the safe operating limit in less than 5 hr.

During the tests, the oil level in the crankcase of the engine was observed by means of a gage glass located on the outside of the engine. If the oil level fell to the bottom of the glass, the test was terminated. The oil consumption, in pounds, during the test was determined by weighing the oil put in before the test and again weighing all that could be drained from the system at the conclusion of the test.

Following the termination of the test, due to either oil consumption or completion of the endurance period, the cylinder and piston were removed for inspection.

The oils herein discussed are shown in Table 1.

Table 2 shows the laboratory analyses of the used oils after each had run its high-power test period.

Table 3 is a tabulation of the running conditions during each test; Table 4 shows the mechanical condition of the piston and rings following the tests.

Table 1 – New Oil Analyses

Oil	A	B	C	D	E
Gravity, deg API at 60 F	27.4	28.0	27.9	27.9	25.6
Viscosity, S.U.S. at 100 F	1600	1575	1680	1548	1980
Viscosity, S.U.S. at 210 F	122.8	120	118.9	120.6	122.4
V. I.	104	102	95.5	104	86.4
Neutralization No. (mg KOH per g of oil)	0.08	0.008	0.03	0.03	0.04
Carbon Residue, %	0.68	0.66	0.30	0.41	0.78
Flash Point, F	520	520	525	550	520
Fire Point, F	580	580	595	610	600
Pour Point, F	+15	+20	+5	+5	+15

Table 2 – Used Oil Analyses

Oil	A	B	C	D	E
Gravity, deg API at 60 F	26.3	26.5	26.1	27.3	24.6
Viscosity, S.U.S. at 100 F	2060	1995	2045	1710	2210
Viscosity, S.U.S. at 210 F	140.3	140.1	132.1	126.8	130.8
V. I.	102	103	95.0	104.9	87.3
Neutralization No. (mg KOH per g of oil)	0.45	0.12	0.68	0.15	0.15
Carbon Residue, %	1.49	1.69	1.13	1.15	1.29
Sludge, % by weight	0.04	0.03	0.11	0.14	0.37
Asphaltenes, % by weight	0.00	0.00	0.04	0.00	0.06
Viscosity Increase, S.U.S. at 210 F	17.5	20.1	13.2	6.2	8.4
Carbon Residue Increase, %	0.81	1.03	0.83	0.74	0.51
Neutralization No. Increase	0.37	0.112	0.65	0.12	0.11

Table 3 – Average Running Conditions

Oil	A	B	C	D	E
Hours at Take-Off	5:00	5:00	5:00	3:15	2:45
Oil Consumed, lb	0.75	1.0	3.5	4.45	4.88
Average Hot Head Temperature, F	395	405	405	405	405
Average Hot Base Temperature, F	410	415	395	390	405
Average Cold Base Temperature, F	385	400	375	380	385
Average Hot Mid-Barrel Temperature, F	500	520	490	485	505
Average Cold Mid-Barrel Temperature, F	455	460	435	435	445
Average Cylinder Oil Jet Flow, lb per min	1.48	1.58	1.47	1.40	1.47

By comparing Tables 1, 2, and 4, it can be clearly seen that the only sound basis for rating the relative lubricating abilities of the oils is the condition of the engine parts following the test. No definite trend of change can be seen in the laboratory analysis results on the oils, either new or following the test period, that would rank the oils in the order of their lubricating ability as shown by the engine parts.

■ Oils Rated by Test Results

Oil A, which furnished the best lubrication of the five oils considered here, has been used in production tests with satisfactory results. During a given one-month period, the piston-ring rejections following the production green-run period amounted to 0.01135 rings per cyl. This figure is quoted on one model of engine which has been in production for some time, and which is considered to be typical of present day production. During the one-month period immediately succeeding the above, oil B was used in production testing an equivalent number of engines of the same model. The piston ring rejections using oil B during this period were 0.0588 rings per cyl, or about five times as many rejections as when oil A was used.

The results of the tests with all of the oils correlate with the mechanical condition of full-scale engines following 70-hr cyclic endurance "Oil Approval" tests and other tests in experimental engines.

The "Oil Approval" test and other tests run with oil A revealed no piston-ring scuffing that could be traced to the oil. After the "Oil Approval" test, which was run at conditions of power and speed much more severe than normal service operation, the piston-ring wear was found to be light. All rings were smooth and in good condition and the oil consumption was low.

The "Oil Approval" test with oil C resulted in several scuffed rings and general piston-ring wear that was heavier than that encountered with oil A. Other testing and service records show comparable conditions following extended periods of operation.

The "Oil Approval" test of oil D severely scuffed rings on four pistons; and after the test of oil E, all top rings showed medium scuffing.

Fig. 4 is a photograph of a typical portion of the piston

■ Fig. 4 – Piston and rings run with oil A

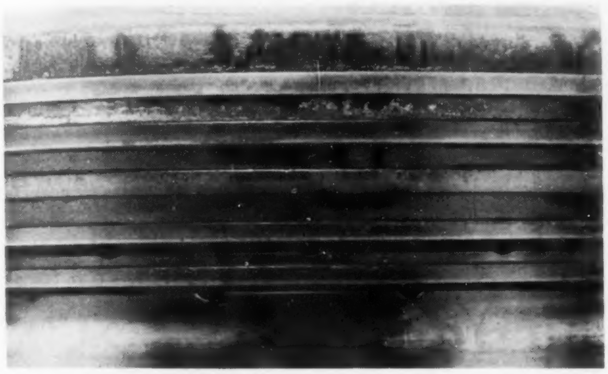
Table 4
Mechanical Condition After Test - Inspection Data

Oil	A	B	C	D	E
Piston Condition	Good Light deposits	Good Light deposits	Good Light deposits	Faces moderately scratched, many short scratches below fifth ring Medium deposits	Thrust face heavily scratched around edges Moderately heavy deposits
Ring Condition, % Taper Removed					
Top Ring	30	60	90—Light-medium scuffed, unevenly worn	100—Heavily scratched, medium scuffed	100 } Heavily scratched and scuffed, sharp and feathered
Second Ring	25	30—Slightly discolored by blowby	30—Lightly scuffed, unevenly worn	100—Medium scratched, unevenly worn	100
Third Ring	25	30	30 } Very lightly scuffed	90 } Lightly scratched	100
Fourth Ring	20	25	30	50	70 } Lightly scuffed, sharp, feathered, unevenly worn
Fifth Ring	20	25	20	50	70
Sixth Ring	20	25	20—Good	40	50—Lightly scuffed, scratched, unevenly worn
	All smooth and in good condition	All very lightly scuffed and scratched		All rings discolored	Top three rings discolored

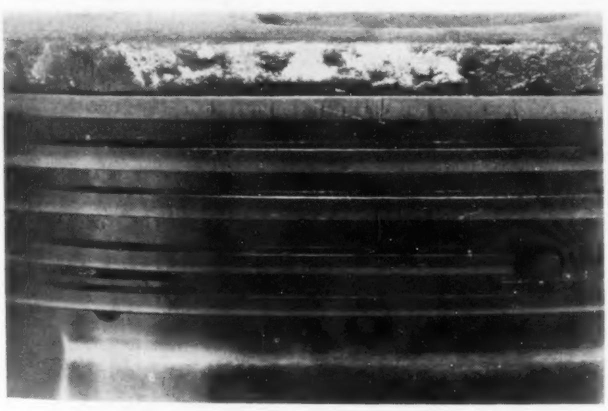
and upper five rings after test using oil A. In this picture, it can be seen that the wear on all rings is light. Although no carbon is present on the face of the top ring, indicating bearing over the entire face, tool marks are visible on the upper 70% of the face. From the presence of these tool marks, the 30% taper removed as recorded in Table 4 was determined. It will be noted that tool marks are visible on all rings, that scuffing marks are totally absent, and that the scratching on the piston is very slight.

Fig. 5 shows the top five rings and section of the piston from the test of oil B. Slight scuffing marks are visible on the top ring and slight scratching on all other rings, but it can be seen that the general wear is light.

Fig. 6, which illustrates the rings and piston from the test of oil C, clearly shows scuff marks on all rings accompanied by some scratching. There are no tool marks visible on the top ring and some very light scratching is evident on the piston.

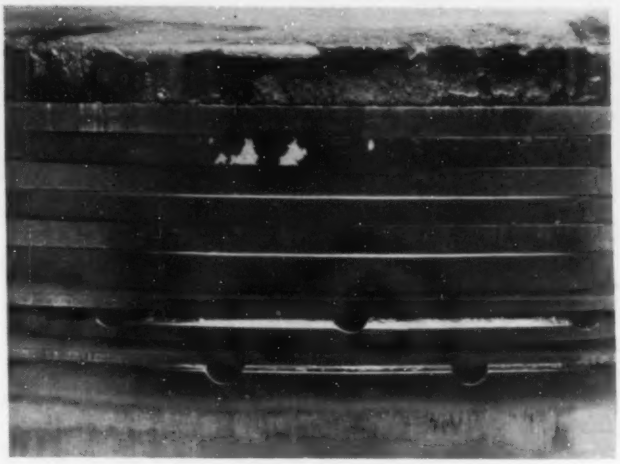


■ Fig. 5 – Piston and rings run with oil B



■ Fig. 6 – Piston and rings run with oil C

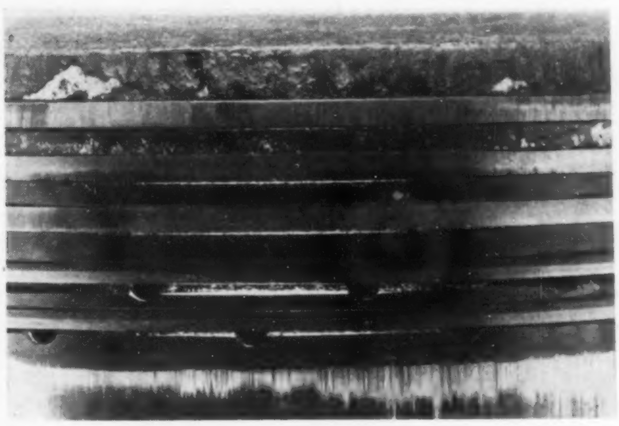
In Fig. 7, the heavy scratching, moderate scuffing, and excessive wear can easily be seen on the top ring. Mainly evident on the second and third rings is the heavy wear, with comparatively light scratching and light scuffing. The oil control rings, fourth and fifth, show heavy wear and scratching. By carefully noticing the width of the bearing face on the latter two rings, the unevenness of the wear can be seen, and the scratching on the piston is clearly discernible below the fifth ring. The flaking of the varnish



■ Fig. 7 – Piston and rings run with oil D

on the second land is indicative of heavier deposits than those shown on the preceding pistons.

As shown in Fig. 8, the piston and rings run with oil E appear very similar to the piston rings from oil D except that the deposits are heavier, as indicated by a greater amount of flaking, the wear and scratching on the rings are generally more severe, and the scratching of the piston face is much greater.



■ Fig. 8 – Piston and rings run with oil E

The data herein presented clearly illustrate that it is possible, by utilizing modern developments in lubricating oils, to supply adequate lubrication to the piston and cylinder wall of the high-output aircooled engine under severe operating conditions.

It is also evident that laboratory inspection is not a sound basis for determination of the lubricating qualities of oils.

In the final analysis, the only satisfactory evidence for comparing the lubricating qualities of different oils is the mechanical condition of the engine parts following testing at standardized conditions. The piston rings offer the most difficult lubrication problem in the engine, and it is therefore consistent that this condition be used as the criterion for rating lubricating oils.

Re-REFINING of Aircraft-Engine Oils

by GILBERT K. BROWER

Materials Engineer, American Airlines, Inc.

IN this paper, Mr. Brower presents data relative to the re-refining of aircraft-engine lubricating oils, based on 12 years' experience with the use of such oils by one of the major U. S. air transport operators.

Stress is placed on this subject at the present time not from the economical viewpoint of peacetime but from a conservation and emergency supply standpoint, which is of prime importance during wartime. A plea is made for more cooperation and coordination, as regards the subject of oil re-refining by the Army Air Forces, the Army's motorized ground forces, the Navy, the WPB, and the OPC, as well as the oil and aviation industries.

THE AUTHOR: GILBERT K. BROWER (M '35), materials engineer, American Airlines, Inc., at present is in charge of the materials laboratory. He has been with American Airlines since 1931, having started with and worked for several predecessor companies including The Aviation Corp., the

THERE are several thoughts which should be kept in mind during this discussion: first, the subject is re-refining – not just simple reclaiming of aircraft-engine oils; second, there is no implied intention that a re-refining process should be attempted on a battlefield[1]: and lastly, such a job *can and should be done* at training bases, maintenance bases, and primary operational points by all military services and commercial operators as well as at the plants of aircraft and aircraft-engine manufacturers, whether they be foreign or domestic.

The information contained in this paper is based on twelve years' experience by one of the major air transport operators in the re-refining of its aircraft-engine lubricating oils. This program was initiated in the latter part of 1930 with the installation of several types of units at Dallas, Newark, Cincinnati, St. Louis, and Chicago. At the present time, all such operations are carried on in one unit installed at New York, since 90% of all oil changes are made there at approximately 100-hr oil change periods.

In this paper, we shall attempt to present, in as concise a form as possible, our experience with several types of oil re-refining units and processes, test data on re-refined aircraft-engine oils, our experience with such oils in actual air transport operations, personnel requirements, as well as personal comments relative to the study of oil conservation.

■ Re-Refining versus Reclamation

Though the expression "reclaimed oil" or "reclamation process" is commonly used for any oil which has been treated to render it fit for further service (particularly by the Air Forces and the oil industry), ~~we feel,~~ very definitely, that a distinction should be made between a reclaimed oil and a re-refined oil. ~~Frankly, we~~ would not use a reclaimed oil in the same manner as one which had been re-refined. It is only too true that much of the adverse criticism directed towards re-refined oils has been based on experience earned by using oils which have been only reclaimed, in the same manner as new oil. We wish, therefore, to point out the distinction we make between these two designations:

By the term, *"reclamation,"* we refer to a process which only removes contaminants such as water and solid substances (insoluble in petroleum naphtha), but not dilution or other products of decomposition. This method is recommended for oils not subject to decomposition or for treating used oils to prolong their useful service life. Such methods as centrifuging, mechanical filtration, and so forth, would come under this heading.

When reference is made to *"re-refining,"* we mean a process which will bring used oils back to a condition substantially equal to new-oil characteristics in all respects. Such a method should preferably provide for clay contact, vacuum distillation, steam stripping, and so on.

[This paper was presented at the National Fuels and Lubricants Meeting of the SAE at Tulsa, Okla., Oct. 22, 1942.]

[1] See *SAE Journal*, Vol. 50, No. 10, Oct., 1942, p. 24.

■ Operation of Re-Refining Units

In this section we shall deal only with the operating characteristics of those units or processes with which we have had personal experience, either from an operating standpoint or from test data obtained in our investigations; code letters will be used for identification of units discussed. No reference is made to other units which we have investigated, but concerning which we have no actual test data. For any given unit or process discussed, the information presented must be considered relative to the dates of our use or investigation.

Unit A – This process consisted of allowing the used oil to settle in drums, then feeding it by gravity to the unit where it went through a vacuum distillation process and was finally permitted to flow by gravity through a specially treated filtering medium, arranged in pans or trays in vertical columns which permitted them to be rotated as they became dirty.

As regards operation of these units, we might point out

that they were entirely satisfactory – very little mechanical trouble, labor cost lowest of any type investigated, and were run at low temperatures due to the vacuum process, with consequently no hazards involved. Oil re-refined by this process, however, would not meet present new-oil specifications. For service conditions in 1930-1932 the oil so treated was approved.

Unit B – In this process, we believe a silicate treatment was used to precipitate sludge and neutralize acidic compounds; this was followed by filtration. We never had occasion to witness personally the detailed operation of

A distinction is made between the terms "reclamation" and "re-refining." A review of service (flight) experience, cost figures, and several re-refining processes is included. A considerable amount of test data on re-refined aircraft-engine oils is presented in order to compare the results obtained by different re-refining units and by slight variations in the process employed by any given unit, and also to point out how a properly re-refined oil is substantially equal to the original new oil.

The test data submitted include not only the conventional routine laboratory tests to determine specification requirements but also aniline points, iodine values, several oxidation (or stability) and bearing corrosion tests.

■

American Airplane and Engine Co., and American Airways at Farmingdale, St. Louis, and Chicago prior to being headquartered at New York. Mr. Brower received his B.A. degree in 1929 from Antioch College, and his M.Sc. degree in 1930 from the University of Pittsburgh.

this unit since it was operated by a subsidiary company. The oil so treated, however, was invariably poor; particularly as regards low flash, presence of water, high carbon content, and neutralization values; even though operated under the supervision of a division engineer. Based on those results, we did not feel that such a process was sufficiently foolproof, and hence no further consideration has been given to it.

Unit C – The following process was never directly operated by ourselves but only by a company which later became a part of our system. We did, however, assemble test data on this process by having some of our used oil re-refined by an independent company using such a method, by obtaining samples from the operator previously operating the unit, and testing samples submitted by another transport operator.

The operation of this process consisted essentially of allowing the oil to flow by gravity through a simple coil-type heat exchanger, thence into a mixing chamber where clay was added. After agitation, the oil and clay mixture was forced by air pressure into a heating still where the temperature was raised to the proper degree, as determined by an automatic thermostat. Steam, supplied by injecting water, was allowed to flow through the oil thus aiding in the elimination of any dilution not removed by heat alone. The vapors driven off passed through an expansion chamber to a condenser and vented receiver. After the steam stripping, the oil and clay mixture was passed into a filter chamber where air pressure forced the clean oil through a paper-type filter, thence through a heat exchanger for cooling, and on into the clean-oil storage tank.

The disadvantages of this method were:

1. Batches were too small, for only 5 gal of oil and clay were mixed at a time. Thus in a 24-hr operation only 200 gal could be produced.

2. Extreme care had to be taken to see that all operating conditions remained substantially the same for all batches in order to make certain that the oil was re-refined to constant specification limits.

3. The filter was of a paper type and poorly designed, which meant that unless the pressure was carefully controlled the filter had a tendency to break. Thus a batch could be completely spoiled and a re-run required.

Unit D – One of the units used with considerable success during the first six years was designed and built by American Airlines' shops. It had a capacity of 500 gal per day.

The process involved was as follows:

Approximately 225 gal of drained oil were pumped into an agitator tank, equipped with closed steam coils, and heated to 275-300 F, during which time it was being kept slowly and continuously agitated. The temperature was maintained until all dilution and moisture had been expelled. At this point flash and viscosity tests were made. If satisfactory, approximately 1 to 1½ lb of fuller's earth was added per gal of oil, and the treatment continued for another 20 to 30 min prior to being filtered. An average loss of 7 to 10% was encountered in this process.

The disadvantages of this unit were:

1. Space required quite large for production capacity.

2. Time factor too long for the time the oil was held at temperature in the open agitator.

3. Digestion with earth occurred at too low a temperature.

4. No satisfactory cooling provisions.

5. Filter process was not foolproof, and hence small quantities of earth could, unless carefully supervised, contaminate the oil.

Unit E – This unit had a production capacity of approximately 150 gal in 18 hr and operated on the following general principle:

The first step in this process was the mixing of the raw stock with an activated earth. The earth and oil mixture was then pumped in the cold state to the cold-oil side of two parallel heat exchangers; one a hot-vapor and the other a hot-oil heat exchanger. The oil then passed to an electrically heated pipe still, where it was raised in a single passage to the predetermined temperature for distillation, that is, 550 F, for the oil being treated at that time. From the pipe still the oil and vapors discharged at their maximum temperatures into a flash tower or separator. The residual oil flowed by gravity to a filter basin carrying with it the clay content. The vapors were passed to the hot-vapor heat exchanger where they traveled counter-current to the incoming old oil, giving up their heat and condensing into liquid form. The residual oil, after going to the filter basin, was drawn by vacuum through a rotary-type filter, which was continually operating, into the hot-oil heat exchanger, imparting some of its heat to the incoming

cold oil and reducing the temperature of the filtered oil. The re-refined oil then passed through a vacuum pump and was expelled on the exhaust stroke as a finished product.

The disadvantages of this process were:

1. A slight cracking reaction occurred, due to high local heat in the pipe still.

2. Some oxidation and possibly polymerization occurred due to filtering at too high a temperature in the presence of air.

3. Better cooling provisions should have been provided.

4. Materials of construction were cheap and in some cases not suitable for the job.

Unit F – This unit was investigated by personally visiting the plant of the manufacturer and having several drums of used oil processed by them for our subsequent study.

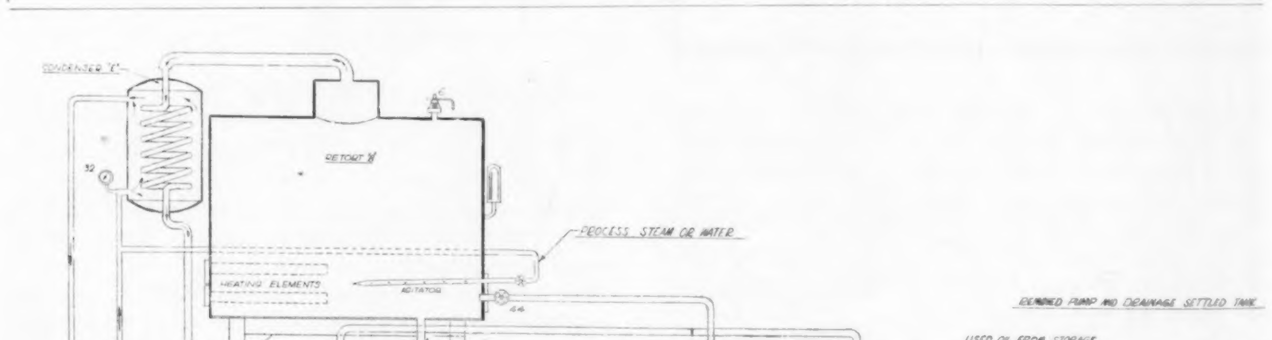
In general, the process used consisted of slowly heating the used oil and a filtering earth under automatic control to evaporate dilution, moisture, and so forth; the distillation products being scavenged by flowing air over the surface of the hot oil, in the case of aircraft oils at a maximum temperature of 425 F. The oil-earth mixture was then dropped from the heating unit to another chamber and forced by air pressure through a filter.

In general there were several advantages to this unit which made it worthy of consideration, such as being of good construction and workmanship, relatively low labor cost involved, available in several sizes. Its main fault, as will be pointed out later, was its inability to meet all new-oil requirements.

Unit G – Refinoil Unit – American Airlines' Process – The unit we are currently using is known as the Refinoil unit and is operated by ourselves under the following general conditions.

Referring to Fig. 1, the used oil is mixed with an activated earth in a mixing tank *A*, and then drawn by vacuum into an electrically heated still or retort *B*, where it is raised, under controlled vacuum and steam stripping, to a sufficiently high temperature to strip off dilution, moisture, and so forth, and maintained at such temperature for a sufficient length of time to insure proper clay contact. After this, the heat and vacuum are turned off, and the oil and earth allowed to flow by gravity through coils (which act as a heat exchanger) in the mixing tank *A* to the contact tank *C*, where a filter aid is added. It is then pumped through a plate- and frame-type filter press *D*, with the filter cloths preceded and/or backed up by a paper filtering medium. The vapors are drawn from the still by vacuum through a condenser *E* to a run-down or distillate tank *F*.

The vacuum used in the process is produced by passing the water, which has acted as the coolant for the condenser, through an inverted-type steam jet. Any type of filtering earth may be used in the contact treatment, such as fuller's earth. We have recently been using two types of activated earths, since by so doing we have only to handle half the amount of material which would be required if raw earth were used, namely, Magnesol and Retrol, with the latter type now being our standard material. For a filter aid, either J-M Hyflo Super Cel or a similar product manufactured by the Dicalite people may be used. Either water injection through a needle valve or direct steam injection may be used to provide the steam required for the stripping operation. No specific data as to temperature should be listed, since this condition is dictated by the grades of oil being re-refined, by the type of earth used, and by the condition of the drainings. We have tried several temperatures, and would say that for most aircraft oils temperatures in the order of 475 to 600 F are satisfactory. Several



■ Fig. 1 – Refinoil machine – flow chart

trial runs at the time the unit is installed and control tests for viscosity, flash, fire, emulsion, gravity and so forth will quickly determine the optimum temperature to use. At present, in our operations, we go to a maximum of 525 F, steam stripping from 425 F on up. Too much "puking" would occur if this operation were started at lower temperatures, particularly if there is much moisture in the oil or earth. The vacuum is generally started at a lower point, approximately 350 F, and gradually built up to a maximum as the temperature is increased, using care to prevent any tendency to "puke." No acid treatment is approved.

■ Test Data

In our investigation of re-refined oils, we have had access and reference to a large amount of test data as obtained by Wright Aeronautical Corporation, Sinclair Refining Company, and our own materials laboratory. Particular emphasis has naturally been placed on our present process and unit (Refinoil). For tests run and the results of tests reference should be made to Tables 1 through 8.

■ Discussion of Test Results

Here we shall present pertinent comments relative to the tests of used oils re-refined by the several units and processes previously referred to, as well as a considerable number of tests on our present process.

■ Used-Oil Analyses

In order to provide a better picture of the products of decomposition and/or contaminants which cause deterioration of oils in service and which must be removed by re-refining, we attach as Table 1 of this report some typical analyses of used aircraft-engine oils.

Note the total sludge content of approximately 1 to 1.5%, of which approximately 15% is tarry matter, 45% carbon and carbonaceous material, and 40% inorganic matter such as silica, lead compounds, and other metallic material. Generally, the neutralization values are slightly higher and must be corrected; there is seldom any dilution as such unless the Army hopper system is used for cold weather starting or unless the used oil has been drained from the lower-output engines. Due to this factor and because most of our oils are drained from the higher-powered engines, we permit our re-refined oils to have a viscosity maximum at 210 F approximately 5 sec higher than the original new oil currently being used.

Table 1 – Typical Analyses of Used Aircraft-Engine Oils

Sample No.	DD	E	ES1	ES2	ES3	ES4	ES5
Date	5-20-41	5-20-41	—	—	—	—	—
Time Since Overhaul, hr	624	589	650	101	482	302	530
Time Since Oil Change, hr	82:06	98:58	60:39	98:20	88:47	100:49	109:54
Viscosity, SU at 210 F	130	129	130	125	129	132	129
Carbon Residue (Conradson)	2.15	2.15	2.12	2.08	2.11	2.43	1.89
Neutralization No.	0.20	0.15	0.10	0.13	0.10	0.10	0.10
Sludge (Wright Method), %	1.33	1.35	1.41	1.41	1.45	1.92	—
Analysis of Sludge:							
Asphaltenes	0.12	0.19	0.21	0.37	0.22	0.32	—
Carbon and Carbonaceous	0.61	0.63	0.81	0.62	0.71	1.05	—
Inorganic Matter	0.60	0.53	0.39	0.42	0.52	0.55	—

■ Tests of Re-Refined Oils – Units A to E

Table 2 lists typical analyses of oils re-refined by units A through E, as previously discussed. These units were once operated by us but are not now considered satisfactory as regards the finished product. The following general comments hold true in each instance.

Table 2 – Typical Analyses of Re-Refined Oils by Units A through E[1]

Unit	A	B	C	D	E
Date	4-32	4-32	7-33	11-32	5-7-37
Compounded	No	No	No	No	Yes
Viscosity, SU at 210 F	123	124	119	120	113
Viscosity, SU at 100 F	1704	1786	1535	1668	1283
V. I.	—	—	—	—	—
Viscosity Ratio	14.4	14.5	12.9	13.9	11.3
Flash, F (COC)	465	480	455	500	425
Pour, F	10	10	10	10	15
Neutralization No.	0.11	0.14	0.065	0.075	0.284
Carbon (Conradson)	1.36	1.68	1.35	1.07	0.57
Precipitation	None	0.05	None	Trace	None
Emulsion, Water at 180 F	DNP[2]	DNP	O.K.	O.K.	O.K.

[1] These units originally operated by American Airlines but not used or approved after the advent of higher-powered engines.
[2] DNP—Did not pass.

Unit A – The neutralization number is quite high, poor emulsion characteristics are indicated, relatively high carbon content is noted (at this time our new oil was averaging 1.0 to 1.2% Conradson carbon), and the oil would not meet the then current Wright Aeronautical sludge and asphaltene test. On the basis of these results we did not seriously consider this unit of further value, particularly so since newer engines of higher output were shortly coming into service.

Unit B – High carbon content and neutralization number noted, poor emulsion, and some precipitation. Not recommended for use even at that time unless considerably improved.

Unit C – Results were relatively satisfactory as regards the finished product though, as noted in this case, a low flash point was sometimes obtained. This unit was not recommended for further consideration primarily because of unsatisfactory design and low capacity rather than the quality of the final product.

Unit D – The oil re-refined by this unit was very good and equal to new-oil specifications then in force, except for a quite frequent tendency to show a trace of precipitation in the final product. This was occasioned by a lack of understanding or care on the part of the operator in his control of the filtering process and partially due to needed improvements in the process. The unit was not considered satisfactory for further use because it was custom-built, required too long a period of time for procurement and replacement of parts, and the fact that for future high-output engines there was some question as to the ability of this process to meet all contemplated new-oil specification requirements.

Unit E – Relatively satisfactory results indicated as regards tests of finished product, except for low flash point. This property could only be overcome by change in equipment design. This unit was not considered sturdy enough or sufficiently standardized to recommend its further use.

■ Tests of Re-Refined Oils – Units F and G

Table 3 lists laboratory tests of oils re-refined by units F and G as compared to new oils then being used. These samples were all obtained prior to our purchase of unit G, and were run per the manufacturers' recommendations. Since then, our new oil has been considerably improved and the method of operating unit G has also been altered; these will be discussed later. The analyses in Table 3 should not, therefore, be considered except in the light of our present knowledge. This table has only been included to point out two factors which we believe should seriously be considered in the study of oil re-refining, namely, the effect of acid treatment on the final product and the variation possible in iodine values.

As regards the effect of acid treatment, it is interesting to observe that sample C-2 (acid treated, using approximately 1 pt of 66 deg sulfuric acid to 50 gal of used oil) has a very low iodine number, low color, and poorer Indiana sludge time than the original new oil (sample G-2) or than sample B-2, re-refined but without the acid treatment. In our opinion, this indicates that the used oil so treated has been refined even beyond the point the original new oil was, and consequently has had some of its good lubricating value removed. Furthermore, from test data on other re-refined oils not acid treated, it would appear that such a treatment is not necessary.

Table 3 – Comparative Tests of New and Re-Refined Oils Units F and G by Manufacturers' Processes

Unit	New Oil	G	G	F[1]
Sample No.	G-2	B-2	C-2[3]	—
Date	4-23-37	4-23-37	4-23-37	4-?-38
Compounded	Yes	Yes	Yes	No
Gravity, deg API	27.9	27.7	27.9	28.2
Viscosity, SU at 210 F	115	112	98	122
Viscosity, SU at 100 F	1340	1256	1029	1545
V. I.	108.5	110.2	109.3	105
Flash, F (COC)	500	475	470	510
Fire, F (COC)	575	560	545	599
Pour, F	15	15	10	20
Neutralization No.	0.236	0.214	0.214	0.028
Carbon (Conradson)	0.57	0.63	0.50	0.61
Precipitation	None	None	None	—
Emulsion at 180 F	O.K.	O.K.	O.K.	—
Ash	0.002	0.0	0.0	0.0
Iodine No.	21.8	24.0	15.2	34.2[2]
Aniline Point, C	129	126	121.5	128.0
Saponification No.	1.76	1.76	2.04	0.29
Color	6–	7	4½+	8+
Indiana Oxidation:				
IST, hr	320	302	243	310
Viscosity Rise at 210 F and 50 hr	19.5	31.7	12.8	—
Viscosity Rise at 210 F and 100 hr	48.4	59.7	38.2	56.1

[1] This oil re-refined from improved-type new-oil drainings with same iodine values as column 1 but with higher viscosity, flash, and IST.

[2] Other confirming tests were 34.3, 36.4, 36.5, 35.7, and 36.0.

[3] Stripped at 625 F.

With respect to the second item, that is iodine value, we believe this test is an indication as to whether or not the re-refined oil has been properly treated; that is, whether or not all products of oxidation and/or polymerization have been removed or have not been formed during the re-refining process; also, as to whether or not such an oil has been overtreated or undertreated in certain respects.

In order to clarify the above, let us refer again to the various analyses in Table 3. First, it must be realized that oils from different crudes and/or oils refined by differing processes vary considerably in their iodine values. We know of three oils currently approved for service having approximately the following values: 7-5, 17-18, and 22-26. We believe that if either of these oils had been re-refined in a different manner, it is quite likely they would not have met the approval tests required. By the same token, when such oils are re-refined, we believe the finished products should have substantially the same iodine values as the original oil. If they do not, and they show higher values, then we should suspect the presence of some products which were not present originally in the new oil and which were formed during either the re-refining process or during service operation and not removed during the re-refining process. If lower than the original oil, we should suspect an overtreatment had been given, and question the stability of such an oil.

■ Tests of Re-Refined Oils – Refinoil Unit

The test results discussed in this section are valid for our present re-refining operations and new oils. Stress is placed here not only on the commonly specified tests but also on several tests indicative of the stability characteristics of the oils tested.

■ Routine Laboratory Tests – Refinoil Unit

First to be considered are the normal routine tests. Referring to Table 4, the most pertinent point to be noted and remembered is that in all instances the re-refined oils are equal to the new oils, even with respect to iodine values and aniline points. The only possible question might be the emulsion results of sample RC-2-08. In this connection, we might say that these test results are most unusual, as in over 259 other similar tests only two have failed to pass the emulsion test with water at 180 F. One instance was traced directly to external contamination, and we feel the same holds true in this instance.

Table 4 – Laboratory Tests Comparison of New Oil versus Re-Refined Oils[1]

Sample No.	New Oil (9410)	RC-2-08[2]	RC-2-11[2]	New Oil (F-392)	R-22
Compounded	Yes	Yes	Yes	No	No
Gravity, deg API	28.2	27.9	27.9	28.3	28.2
Flash, F (COC)	525	510	530	540	520
Fire, F (COC)	595	600	600	610	600
Viscosity, SU at 210 F	122.0	121.6	121.6	122.5	123.2
Viscosity, SU at 100 F	1533	1505	1519	1562	1568
V. I.	106	107	106	105	105.2
Pour, F	5	15	15	5	15
Carbon (Conradson)	0.624	0.628	0.617	0.57	0.595
Ash	0.000	0.008	0.008	0.000	0.000
Neutralization No.	0.10	0.075	0.10	0.05	0.075
Saponification No.	1.1	0.45	0.34	0.11	0.28
Color	6–	7+(5–)	7–(5–)	4½	6++(4+)
Color Stability (350 F for 45 min)	6+(4+)	8–(5–)	8–(5–)	8–(4½+)	8(4½++)
Iodine No.	22.4	22.8	22.2	22.4	22.7
Aniline Point, C	128.5	127.0	126.0	128.5	125.0
Sulfur, %	0.085	0.097	0.095	0.07	—
Navy Emulsion—NaOH	—	Ng76 Ng69	Ng4 OK	—	—
—NaCL	—	Ng Ng	Ng OK	—	—
—Water	—	Ng Ng	OK OK	—	—

[1] Refinoil unit—American Airlines process.

[2] Different activated earths used.

■ Indiana Oxidation Tests – Refinoil Unit

Table 5 is a compilation of Indiana oxidation tests (not Stirring) run not only under the normal standard conditions,

that is, in glass and at 341 F, but also under more severe conditions such as in an iron tube to act as a catalyst and at 450 F for its accelerated effect.

These tests indicate that from an overall standpoint the new oil has a slight edge, though in several instances the re-refined oils are superior. The most obvious point in which the re-refined oils do not come up to the new oils is the increase in viscosity at 210 F and 100 hr.

Table 5 – Indiana Oxidation Tests
Comparison of New Oil versus Re-Refined Oil [1]

Sample No.	New Oil (9410)	RC-2-08[2]	RC-2-11[2]	New Oil (F-392)	R-22
Compounded	Yes	Yes	Yes	No	No
In Glass at 341 F:					
IST, hr (10 mg per 10 g)	450	464	397	470	407 408
1.0% Naphtha Insoluble, hr	496	497	445	499	433 435
Viscosity Rise at 210 F and 50 hr	11.7	5.6	11.6	10.5	15.8 15.8
Viscosity Rise at 210 F and 100 hr	28.0	39.0	33.4	29.1	38.0 39.0
In Iron Tube at 341 F:					
IST, hr	326	358	394		
1.0% Naphtha Insoluble, hr	355	386	437		
Viscosity Rise at 210 F and 50 hr	19.0	27.4	16.4		
Viscosity Rise at 210 F and 100 hr	48.6	72.0	45.4		
In Glass at 450 F:					
IST, hr	137 130	104 103	126		
1.0% Naphtha Insoluble, hr	158 140	117 115	140		
Viscosity Rise at 210 F and 50 hr	35.2 47.6	40.2 40.6	42.2		
Viscosity Rise at 210 F and 100 hr	91.0 137	213 230	170		
In Iron Tube at 450 F:					
IST, hr	110	130 102	103		
1.0% Naphtha Insoluble, hr	127	151 113	135		
Viscosity Rise at 210 F and 50 hr	31	29.6 29.4	36.4		
Viscosity Rise at 210 F and 100 hr	87.2	126 140	135.6		

[1] Refinoil unit—American Airlines process.
[2] Different activated earths used.

■ Underwood and Oxygen Absorption Tests

In Table 6 oxygen absorption values and Underwood oxidation test data are tabulated.

The oxygen absorption values, indicating the minutes required for absorption of 3000 cc of oxygen at 400 F, show the compounded new oil to be considerably more stable than either of the two re-refined and re-compounded oils; however, the new and re-refined non-compounded oils are rated about equal.

With respect to the Underwood oxidation tests, reference should particularly be made to the low loss in milligrams of cadmium-silver by the re-refined oils compared to the new oil: only 50 to 60% of the new oil loss. As regards viscosity rise, the new oil has the edge, but when sludge and asphaltenes are compared both new and re-refined oils are quite low.

Table 7 – Sinclair's Bus Engine Sludging Tests [1]
Comparison of New Oil versus Re-Refined Oil [2]

Sample No.	New Oil (9410)	RC-2-08[3]	RC-2-11[3]	New Oil (F-392)	R-22
Compounded	Yes	Yes	Yes	No	No
Lead Bath Roof Deposit at 150 hr.	20.7	18.6	8.3	21.7	22.9
General Condition of Roof	Coked	Coked	Coked	80% Coke	85% Coke 15% Lacquer
General Condition of Sump	Clean	Clean	Clean	Clean	Clean
Used Oil Analyses at 150 hr:					
Tar, %	0.488	0.408	0.544	3.792	1.32
Asphaltenes, %	0.296	0.248	0.344	1.336	0.44
Chloroform Insoluble, %	0.192	0.160	0.200	2.456	0.88
Ash, %	0.000	0.028	0.000	0.000	0.000
Viscosity, SU at 210 F	1052	1302	778	1897	2032
Carbon (Conradson)	5.031	5.370	4.383	5.045	5.22
Ash	0.135	0.026	0.081	0.213	0.38
Neutralization No. (Method A)	5.8	12.2	10.0	19.1	3.6
Viscosity Rise at 210 F and 150 hr	930	1180	656	1773	2009
Carbon Rise at 210 F and 150 hr	4.407	4.734	3.766	5.41	4.82

[1] Refer to Table 8 for bearing corrosion data.
[2] Refinoil unit—American Airlines process.
[3] Different activated earths used.

Results of another stability test, known as the Bus Engine Sludging Test, are tabulated in Table 7.

It will be noted from these results that the re-refined oils are substantially equal to the new oils in all respects – in some instances they are slightly better, in others, a little poorer. Bearing corrosion data on this test are included in the next section.

■ Bearing Corrosion Tests – Refinoil Unit

Table 8 is based on a compilation of several bearing corrosion tests. Any comments relative to these tests should be prefaced by the fact that in service there was never any difference noted, as regards bearing corrosion, between engines operated on new oil only and engines operated on a mixture of new and re-refined oils. We present these figures for what they may be worth. It seems to us that a considerable amount of study still needs to be done towards arriving at a satisfactory method for determining bearing corrosion which will to some extent be related to service

Table 6 – Underwood Oxidation and Oxygen Absorption Tests
Comparison of New Oil versus Re-Refined Oil [1]

Underwood Oxidation Tests: Sample No.	New Oil (9410)			RC-2-08[2]			RC-2-11[2]		
Compounded	Yes			Yes			Yes		
	Original Oil	0.01% Fe_2O_3 Added 5 hr	0.01% Fe_2O_3 Added 10 hr	Original Oil	0.01% Fe_2O_3 Added 5 hr	0.01% Fe_2O_3 Added 10 hr	Original Oil	0.01% Fe_2O_3 Added 5 hr	0.01% Fe_2O_3 Added 10 hr
Cadmium-Silver Loss, mg	—	804	1528	—	495	984	—	464	736
Viscosity, SU at 210 F	123	280	542	119	290	891	122	259	612
Carbon (Conradson)	0.69	2.96	4.24	0.68	3.19	5.41	0.68	2.99	4.65
Neutralization No. (Method A)	0.10	—	—	0.13	—	—	0.10	—	—
Neutralization No. (Method B)	—	5.65	7.9	—	2.25	9.10	—	5.10	11.10
Naphtha Insoluble, %	—	0.02	0.03	—	0.02	0.04	—	0.02	0.05
Chloroform Insoluble, %	—	0.01	0.01	—	0.01	0.01	—	0.01	0.02

Oxygen Absorption Tests (pure O_2 at 400 F): Sample No.	New Oil (9410)	RC-2-08[2]	RC-2-11[2]	New Oil (F-392)	R-22
Compounded	Yes	Yes	Yes	No	No
Min for First 3000 cc O_2	87	47	69	29.9	27.0

[1] Refinoil unit—American Airlines process.
[2] Different activated earths used.

Table 8 – Bearing Corrosion Tests
Comparison of New versus Re-Refined Oil [1]

Sample No.	New Oil (9410)					RC-2-08[3]					RC-2-11[3]					New Oil (F-392)					R-22				
Compounded	Yes					Yes					Yes					No					No				
BEST Bearing Corrosion Tests[2]:																									
Hr	40		75		150	40		75		150	40		75		150	40		75		150	40		75		150
Copper-lead	335		652		1261	566		1129		1845	—		—		—	—		543		1326	215		617		1312
P & W	37		65		127	31		68		118	26		50		75	—		616		1399	174		602		1323
WAC-Lead Flash	103		221		431	163		324		448	149		251		469	61		317		375	3		35		98
WAC-Lead Flash & Indium	6		18		50	15		31		54	11		50		168	88		180		417	40		142		326
Chrysler Bearing Machine:																									
Hr	5	10	40	75	89½	5	10	15	20	33	5	10	40	75	79½	5	10	40	75	83	5	10	40	72	
Loss, mg	5	12	50	106	184	5	20	60	125	927	5	12	50	105	150	8	12	23	157	310	12	20	50	300	
Underwood Oxidation Test – Bearing Corrosion Loss (0.01% Fe_2O_3 Added)[4]:																									
Hr	5	10				5	10				5	10													
Cadmium-Silver Loss, mg	804	1528				495	994				464	736													

[1] Refinoil Unit—American Airlines Process.
[2] Refer to Table 7 for other tests.
[3] Different activated earths used.
[4] Refer to Table 6 for other data.

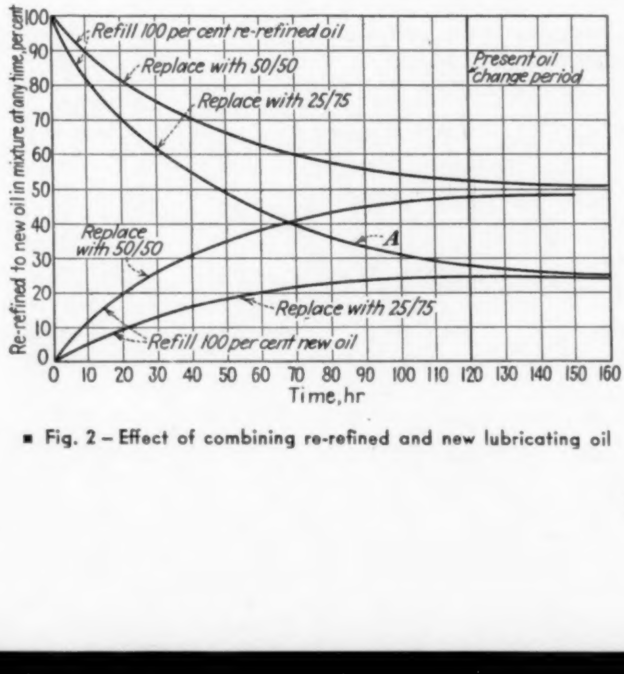
experience. It is to be regretted that we have no data based on the MacCoull-Ryder machine as operated by experienced personnel. We hope to overcome this missing detail in the near future.

In reviewing the bearing corrosion data available, a considerable variation is noted. For instance, sample RC-2-08 shows up very poorly on the copper-lead loss both by the BEST test and the Chrysler Bearing Machine. The same sample is very good as regards the cadmium-silver loss on the Underwood test. Both the re-refined oil sample R-22 and the new oil sample F-392 show up badly on the BEST test on a Pratt & Whitney bearing, while with re-refined oil sample R-22, in the same test but on the Wright lead flash-plated bearing, the loss was considerably less than any other new or re-refined oil samples.

No actual flight tests have been made specifically to compare the operating experiences of an engine run on re-refined oil versus new oils, due to the fact that re-refined oils are not used in such a manner in service.

Our flight experience has been based on using re-refined oils, as produced at the base stations, in the same manner as new oil would be used, for refills and make-up. Most conclusive in this respect was the use of over 5000 gal of re-refined oil per month in our aircraft engines operating out of New York during the period from August, 1940, through May, 1941.

Fig. 2 shows the effect of combining re-refined and new lubricating oil during normal operations on the basis of varying percentage combinations of re-refined and new oils. Probably that indicated by curve *A* would be most indicative of our present service.



■ Fig. 2 – Effect of combining re-refined and new lubricating oil

In an attempt to give some substance to operating costs of re-refining operations, we have included Table 9.

These figures are based on our operations when using a new oil containing an additive, hence are slightly more than 5¢ per gal higher than if a straight mineral oil was so treated.

As a matter of interest, we might add the following:

We have never approached our maintenance allowance; our depreciation figure is based on a five-year period when operating at approximately half of the possible production capacity of the unit; but the labor cost may be increased by 1 to 2¢ per gal during these times.

Table 9 – Typical Re-Refining Cost Data
Refinoil Unit – American Airlines Process

	c per gal
10% Concentrate	5.55
Earth (Retrol)	1.47
Filter Aid (Johns-Manville Hyflo Super Cel)	0.46
Filter Paper	0.34
Labor	4.18
Depreciation	1.00
Maintenance	1.00
Electric (estimated)	1.00
Total Cost	15.00

One of the questions which may logically be raised is that regarding the possible quantities of used oil available for re-refining. This is naturally not an easy question for a layman to answer without access to a considerable amount of data now considered confidential. We can say this much: in normal air transport operations as carried on during the past four years, a saving of 25% in total oil purchased is possible. This percentage is based on operating twin-motored equipment, with oil change periods of 100 hr, with 90% of the oil changes being made at a major base, and considering oil consumption at 0.5 gal per hr per engine. For example, if total oil purchased amounted to 500,000 gal, the amount actually consumed would be approximately 350,000 gal and the amount drained approximately 150,000 gal; assuming 90% recovery in the re-refining of these drainings, the amount re-refined would equal 135,000 gal resulting in an approximate conservation or saving of 25%.

As regards the amounts capable of being saved by the

aircraft-engine and aircraft manufacturers, we can only say that a considerably higher percentage could be conserved by re-refining, say conservatively in the neighborhood of 40 to 50%. This is true because they make very frequent oil changes and the actual consumption in relation to amount drained is low in comparison to air transport operations.

Referring to the military services, we naturally do not have any definite figures available. Consider, however, the above mentioned possibilities and the fact that at training bases the flight time versus oil change periods is no doubt somewhere in between air transport operations and the manufacturer's procedure; that at major maintenance and primary operational points there should not be too much difference from air transport operations, except perhaps for more frequent oil change periods as a safety precaution; then you might realize that a considerable quantity of good oil is now utterly wasted which could be conserved by re-refining. Naturally, airplanes at dispersed points present a different problem, and we do not know how the percentage of such planes compares to the total.

For all types of operations, a figure of 10% seems conservative.

■ Discussion of Oil Conservation

Though this paper has dealt primarily with aircraft-engine lubricating oils, the same story could be applied to any heavy-duty automotive lubricating oil. We would like, therefore, to add the following personal comments:

After being in the war for nearly a year, and preparations having been going on for over two years, there is still apparently a sad lack of appreciation as to how important oil is in this war.

Approved re-refining units should be available at all key bases; or use should be made – after thorough investigation naturally – of plants locally available within 75 to 125 miles of the subject locations. During the last war most of our Air Corps bases considered it worthwhile to have small reclaiming units and test facilities.

We believe one of the bases has recently had two different types of re-refining units installed; but we do not know that anyone is closely following the actual operation of these units either through laboratory tests to determine compliance with new-oil specifications or through engine inspections to determine satisfactory service performance.

A good deal of the cause for this condition can be placed squarely in the lap of the oil industry because of the advice certain of their representatives have given when called upon in a consulting capacity.

The following conclusions and recommendations are based on our personal experience in studying this subject for many years and, if adopted as a unified program – preferably by the Army Air Forces, Quartermaster Corps, Ordnance Department, WPB, OPC, and others, or at least by some one qualified group in each branch – they would result in definite answers within a very few months.

■ Conclusions

A. Crankcase oils drained from automotive equipment – ground or aircraft – operated by the Army Air Forces, Navy Quartermaster Corps, or Ordnance Department may readily be re-refined to new-oil specifications; providing a proper type unit and a correct process are employed. A job as thorough as this should only be done at base and reserve depots. If required, additives may satisfactorily be added to meet the original new-oil specifications. In some, though few, domestic localities, advantage could be taken of now existing commercial re-refining plants.

B. For treating drained crankcase oils in the field, if desired, no attempt should be made to re-refine the oil, but rather a thorough job of reclamation only should be done. This would be more in the nature of extending oil drain periods and might be accomplished by using small portable equipment such as a centrifuge or filtration units which would not remove any additive.

C. Specialized personnel are not required, only men with good common sense, plus approximately three weeks training.

■ Recommendations

Immediate steps should be taken to obtain test data on small practicable re-refining units currently manufactured to a standard design and following a well planned program. This program should be flexible enough to obtain as much data in as short a time as possible and should include not only routine laboratory tests but at least two or three stability or oxidation tests of the oil produced from each unit. Essentially, such a program should be along the following line:

A. It should be under the direction of a central control agency or group, as for instance, the National Bureau of Standards or the Cooperative Research Council.

B. A list of reputable manufacturers of standardized equipment for re-refining of oils should be obtained.

C. A sufficient amount of drained crankcase oils, each from a common source, of several different grades and types should be made available in order that each equipment manufacturer would have enough for a normal batch or run of each. For instance, the following types and grades might readily be obtained: non-compounded, Pennsylvania-base aircraft oil; a compounded (any currently used additive) Mid-Continent, solvent treated aircraft oil; a straight Pennsylvania-base motor oil; and a compounded, Mid-Continent base motor oil.

D. A sample of each of these oils should be distributed under code to each equipment manufacturer. A representative of the agency directing the program should then visit each manufacturer's plant, witness the re-refining of each oil, noting in detail the process followed, identifying in code, and sealing each sample produced.

E. The coded samples should then be distributed among several participating laboratories (for instance, the control laboratories of the major oil companies) with a duplicate or retained sample to the control agency, for tests outlined by those conducting the program. Such tests should include all the commonly accepted routine laboratory tests, plus the aniline points and iodine values, and at least two or three of the currently used oxidation or stability tests. One or two laboratories could not conduct these tests in a reasonable time.

F. The test results should then be submitted to the control agency and an analysis group of all interested parties set up to study the test data submitted, the processes involved, and the practicability of the units for the intended services. On the basis of this study an approved list of

satisfactory units and processes should then be prepared for those parties concerned.

G. After the above information has been obtained, proposed methods of control should be considered, that is, for any given service what should the control tests consist of and by whom should they be made.

The author wishes to acknowledge his appreciation to the Wright Aeronautical Corp. and the Sinclair Refining Co. and their personnel who furnished a major part of the test data presented.

Aviation Power Plants

THE integral powerplant plan means that an aviation powerplant would be built to form the front section of a nacelle – designed, assembled, and tested as a unit by groups specializing on this job. These powerplant specialists would arrange the assembly of engines, turbosuperchargers, generators, propellers, and all accessories, to get a certain overall performance, one of the items on which plane performance depends. Other groups of aerodynamic specialists would, in parallel, prepare the plane up to the nacelle fire wall, and provide the studs or fastenings to which the integral powerplant section would be attached. The fire wall might be a part of the powerplant section, with the various accessories bolted on both sides of it. It is to be expected that there might be achieved a standardized arrangement, so that any plane would take any integral powerplant of a certain specification. One might even dare to hope for a British-American standard, so that an American integral powerplant would fit a British plane, and conversely. Of course, there would have to be standards for weight, center of gravity, thrust, shape and arrangement of cowling, and so forth. Aerodynamic specialists can be depended upon to handle the situation that might arise in connection with center of gravity. Perhaps the plan is going to be difficult to execute in the single-engine planes. However, it may be expected that aerodynamic specialists can find some ingenious way to obtain a satisfactory balance without shifting powerplant parts.

To American engineers and manufacturers, who have brought standardization to such a high pitch, the advantages of this integral powerplant plan seem obvious. No doubt the plan originally was started so as to permit of rapid replacement of powerplants or airplane structures in the field, when damaged in war. We in America have not been up against such problems very long, and so perhaps we have not caught up with the integral powerplant scheme in spite of our boasted skill at mass production. But ease of replacement is a major advantage both in war and peace. Rapidity of production so much needed in wartime is now being helped by manufacture of various components in parallel as independent groups, later assembled to give a complete apparatus. The integral powerplant helps this nicely.

The integral powerplant plan will permit improvements to be made with greater ease. A redesigned powerplant section can be used with an existing plane design or a redesigned plane with an existing powerplant.

The writer has been associated intimately with turbosupercharger development, and hopes that the entire turbosupercharger addition to the airplane engine will become a part of the integral powerplant. This will include the turbosupercharger itself, the flexible joint connecting it with the exhaust manifold, the ducts with ramming intake supplying cooling air to turbosupercharger and air cooler, the supercharged air ducts from turbosupercharger to air cooler and air cooler to engine intake manifold, the air cooler and the exhaust ducts. It is to be expected that this will be a marked improvement over many existing designs where the turbosupercharger parts are independent entities with lengthy duct connections.

So far as can be told from current literature, the engine builders have taken the initiative with the integral powerplant plan in both England and Germany. But an opposing idea has been suggested, that the initiative for creating integral powerplant design must remain with the plane builder along with the responsibility.

But in any event, the integral powerplant plan would best be executed by specialized groups devoting themselves exclusively to the design and manufacture of integral powerplants. The designers and manufacturers who are producing modern airplanes have done a wonderful job, but the complications are getting greater and greater. The integral powerplant plan would advance the arrangement, which already exists to some extent, of specialization by independent experts, on design and manufacture for (A) aerodynamic design and construction; (B) airplane engines; (C) turbosuperchargers; (D) propellers; (E) generators and other airplane accessories; (F) assembly of B, C, D, and E as the integral powerplant.

Such specialization already has come about in the design of steam powerplants, and specialists design and erect an entire powerplant, including boilers, engines, and all accessories. Exigencies of war now have introduced a similar plan so far as manufacture only of one aviation powerplant is concerned, and all powerplants for a certain type of plane now are being assembled and tested in a single factory and supplied to various other manufacturers among whom the production of the plane has been divided.

The execution of the proposed plan may result in cutting a shoot from the parent aeronautical stem and starting to grow it as a separate integral powerplant tree. An airplane powerplant certainly is automotive apparatus, so the Society of Automotive Engineers might attempt fertilization of this new shoot. This might hold, no matter how the specialized powerplant groups were organized. As science and engineering have progressed in the past century or so, engineering groups often have been subdivided in this way. Once there were only military and civil engineers. As engineering complexity increased, there have budded off from the parent stem of civil engineering, shoots which have grown into mining engineers, mechanical engineers, electrical engineers, and recently, steam powerplant engineers. Engineering of water-ships (we now have to say it this way) is divided between marine engineers and naval architects. The rapid increase of complexity of the airplane from the Kittyhawk model of 1903 would seem to demand similar specialization of the aerodynamic and the powerplant groups.

Excerpts from the paper of the same title by Sanford A. Moss, General Electric Co., presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 12, 1943.

FLIGHT-TESTING EQUIPMENT for Large AIRCRAFT

by W. T. DICKINSON
Douglas Aircraft Co., Inc.

THE first of the modern flight-testing-equipment installations was made in the Douglas DC-4E. In the tests of this airplane, Mr. Dickinson says, all data were recorded manually, except that two cameras were used for some special tests. The test equipment was so extensive that the flight crew always had from 12 to 22 men. The undesirability of carrying such a large crew, the necessity for continuously recorded data, and the fact that the need for data is often not sensed until after the test, made it desirable to develop some sort of mechanical recording means to take over as much of the recording job as possible.

The flight-testing equipment for testing modern large airplanes, such as the XB-19, has been developed to the point where the small crews now required do not need to record any of the data. Back in the sleeping compartment, completely unattended, automatic flight recorders and an automatic temperature recorder system continuously gather all possible information on the operation of the engines, controls, and performance of the airplane.

To take care of the flight-test equipment, it is necessary to design the interior specially. This interior protects the airplane and provides solid anchorage for every piece of equipment and ballast that might come loose in maneuvers.

All in all, the test instruments, interior furnishings, wiring, tubing, batteries, and electrical accessories for one installation weighed nearly 11,000 lb and cost $38,600, exclusive of the cost of labor to install.

THE AUTHOR: W. T. DICKINSON has held various positions with the Douglas Aircraft Co., Inc., since he joined the company in 1934. He has been head of the Engineering Flight Research Group of the Douglas Santa Monica Plant since 1939. In this capacity he has charge of the engineering flight tests of the Douglas Models DC-3, B-23, A-20C, B-19, C-54, and other related models. Mr. Dickinson is a graduate of the Massachusetts Institute of Technology with an S. B. degree in aeronautical engineering.

WHEN the XB-19 took to the air on its first flight on June 27, 1941, the small crew on board were completely occupied by their duties, much too occupied to record any useful engineering data. However, back in the sleeping compartment, completely unattended, an automatic flight recorder and automatic temperature recorder were continuously gathering in all possible information on the operation of the engines, controls, and performance of the airplane. In view of the large insurance costs involved, a few engineering wags charted the first minute in the life of the XB-19, which is shown in Figs. 1A and 1B.

Flight-testing equipment has improved in quality and scope steadily as the aeronautical art has advanced. The first of the modern flight-test-equipment installations was made in the Douglas Model DC-4E, which airplane was scheduled for airline service, and it was intended to find out as much as possible about the airplane in as short a time as possible.

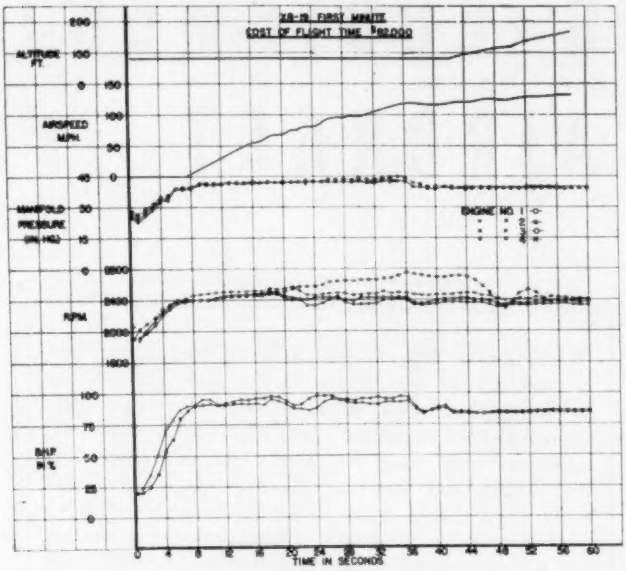
In this airplane all data on all flights were recorded manually, except for one 16-mm movie camera and one Leica, which were used for some special control-system tests and wing-drag studies. Because the test equipment was extensive, the required flight crews always numbered from 12 to 22 men. In addition to large crews, which were considered undesirable, problems were frequently arising that required continuously recorded data, often under unfavorable human conditions. Also, in many cases it occurred that the trend of events leading up to some particular test would have been useful but the information was not available because its importance was not sensed at the time, and the data were not recorded. For these reasons the Douglas Co. chose to eliminate manual recording of data and to concentrate on the development of mechanical recording means wherever possible. To this end we have worked for the past three years.

It is the purpose of this paper to describe briefly the flight-test equipment currently being used by the Douglas Co. We also hope to show something of the scope of the problem at hand when considering the equipment required for adequately flight testing modern large aircraft such as commercial airliners or cargo planes.

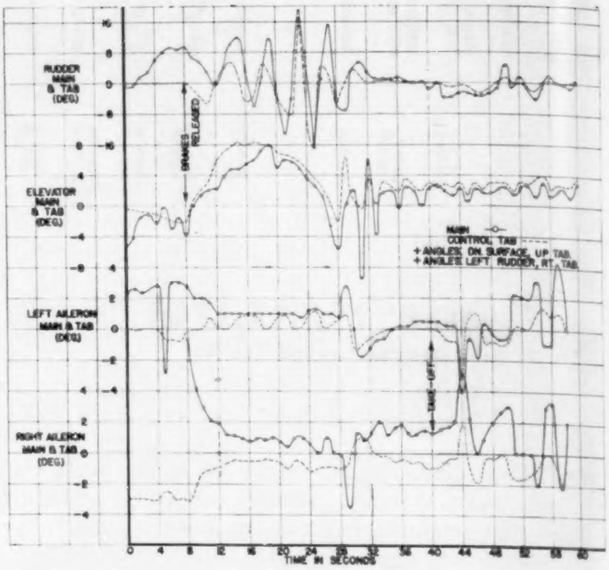
Flight Recorders

Flight recorder is the title that we have arbitrarily given to an assembly consisting of a camera, instrument panel, timing unit, relay box, and remote control box. We have adopted a camera method as standard, because it allows the

[This paper was presented at the National Aircraft Production Meeting of the SAE, Los Angeles, Calif., Oct. 1, 1942.]



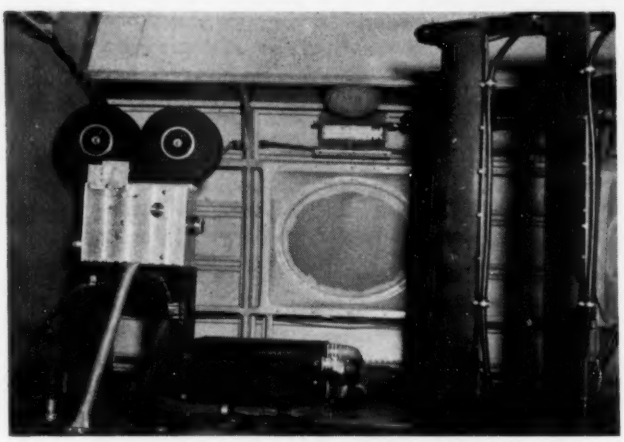
■ Fig. 1A – First minute of flight of the XB-19 – various operating conditions



■ Fig. 1B – First minute of flight of the XB-19 – operation of control surfaces

use of standard instruments, of which we expect and receive a high order of reliability and accuracy.

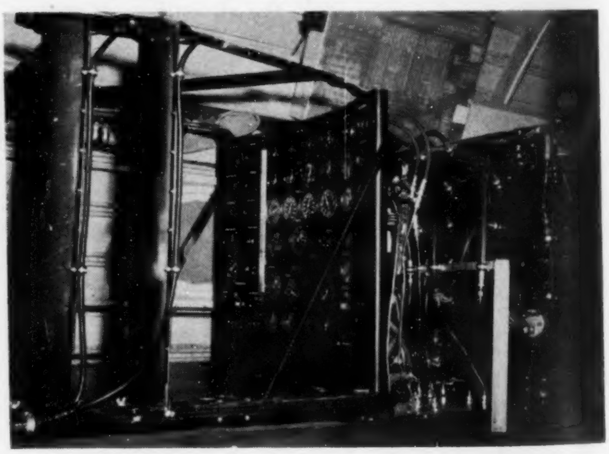
Several different methods of utilizing a camera for flight-test work have been in service for many years. However, our latest design, shown in Fig. 2, is completely flexible in that it can take pictures at convenient intervals from one every 30 sec to 24 frames per sec. The figure shows the camera on its pedestal with a 400-ft film magazine installed and the timing unit in front.



■ Fig. 2 – Main flight recorder camera installation

The camera itself is a reworked De Vry movie camera incorporating a powerful magnet that is actuated by impulses received from the timer for all single-shot type of operation. The camera is also provided with a detachable governor-controlled ciné drive motor, which can be cut in for continuous operation at any speed from 4 to 24 frames per sec. For use with external magazines of either 400 ft or 1000 ft, a suitable take-up motor is mounted on the camera.

Fig. 3 illustrates a typical flight recorder instrument panel installation that was recently employed during the flight tests of the Douglas Model C-54. This is what we



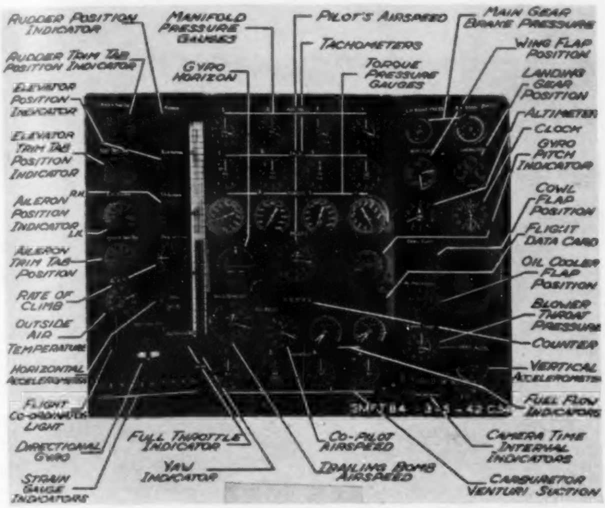
■ Fig. 3 – Main flight recorder panel installation

call a "large" panel: its dimensions are 24 x 32 in. It is mounted on a rigid platform together with the camera shown in Fig. 2, and the whole unit is then shock mounted to a frame built up from the floor. Four adjustable 40-w fluorescent lights are used, and may be seen in Figs. 2 and 3.

A full view of this, the main flight recorder panel, is shown in Fig. 4. Most of the instruments in the panel are familiar to all. Some, however, are of special interest.

The manometer shown on the panel is a yawmeter. It obtains its pressures through two holes located identically

on either side of the nose on the centerline of the ship and about 18 in. aft. Care was taken to get the holes lined up identically on both sides, and the flight test results were very satisfactory. This unit is so light and easy to install (on ships with no engine in the fuselage) that it is also recommended for airline cruising operations to promote more accurate flying.



■ Fig. 4 – Full view of the main flight recorder panel

The gyro pitch indicator is a Sperry-built unit and has a mechanism similar to an artificial horizon except that the pitch indications are magnified and extended to ± 20 deg. The indicator is installed in the ship so that the 0-deg mark coincides with the fore-and-aft centerline of the ship, thus making measurement of the wing angle of attack possible.

The vertical accelerometer is a hydraulic-type accelerometer made from a suitable liquid column and a sensitive pressure gage. The advantage of this type of accelerometer is that it is highly accurate, it is easy to build, and its components can be altered at will to give full-scale deflection for any desired maximum acceleration.

The fore-and-aft horizontal accelerometer is located close to the airplane center of gravity and parallel to the ship's centerline. The accelerometer used is the standard Douglas variable-reluctance type recording on the voltmeter shown in the picture. This unit has already been completely described[1]. For this particular use, the sensitivity of the accelerometer was increased to a full-scale deflection of about ± 0.5 g. It should be noted that a change in the axis of the accelerometer causes a zero shift, and this effect is measured by reference to the gyro pitch indicator.

The lights numbered 1 to 10 in the lower left-hand corner (Fig. 4) are the lights that designate which strain-gage run is in operation. The lights in the lower right-hand corner show the camera time interval in use, and the light to the left of the directional gyro is the take-off and landing coordinating light, which will be described later.

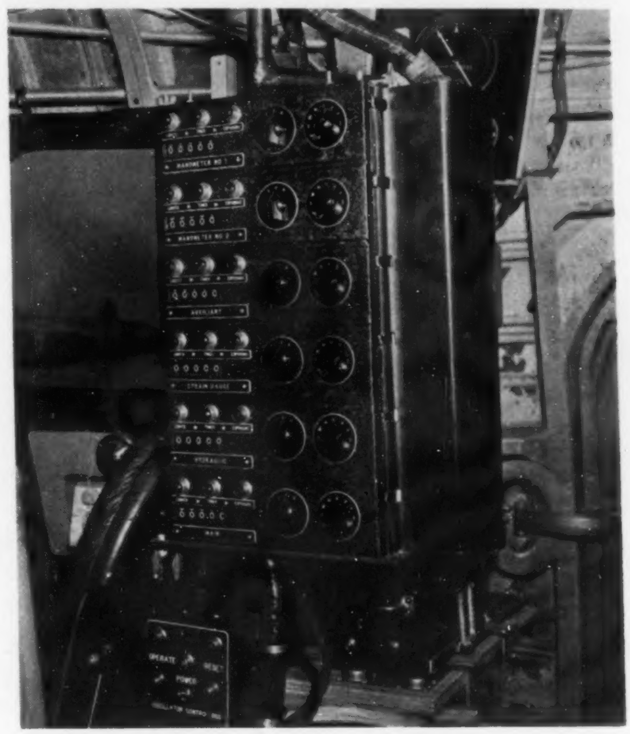
[1]See *Instruments*, Vol. 12, No. 12, Dec., 1939, pp. 324-326: "New Applications of Variable-Reluctance Method to Aircraft Instruments," by H. H Bruderlin

The timing unit, which can be seen in front of the camera in Fig. 2, is designed to provide electrical impulses for the camera operation at 0.5-, 1-, 2-, 5-, 10-, or 30-sec intervals. The timing interval is selected at the control box.

The unit is ingeniously simple, consisting essentially of a governor-controlled motor operating a series of micro switches through a system of gears and cams. Each cam is properly lobed to give the desired timing interval, and each cam follower is a roller supported on the micro switch. Thus the time-interval selector through the relay box determines which cam is in use, and the micro switch completes the circuit when the roller falls into the troughs of the cam. The impulses are fed through a separate cable to the remote control box and thence to the flight recorders.

The relay box contains the relays necessary for the operation of the system and also serves as the junction box for each flight recorder.

The last remaining unit required to complete the flight recorder system is the control unit. Fig. 5 shows the installation of six control boxes, one for each flight recorder in the ship. The stack of control boxes is located in the pilot's compartment at the chief flight engineer's station.



■ Fig. 5 – Flight recorder controls and take-off and landing co-ordinator controls

All the control boxes are identical and can be racked up as shown or fastened separately if desired. The control wiring circuits are arranged in such a way that even when any number of units are connected in, complete flexibility is still available as follows:

(a) Each flight recorder can be operated independently, or

(b) Any one box can be selected as the master control, and all the flight recorders will then operate from this box only, or

(c) Any combination of control boxes can be operated from a master control and the rest operated independently.

Each control box contains a counter that is impulsed by the operating current going to its flight recorder camera; thus the number of exposures taken by each camera can be watched. For long flights this feature has been found very desirable.

The flight recorder system described above has been in use on our flight tests for some time. On the Model C-54 alone, over 26,000 ft of flight-test film were taken on this system.

The cameras have been severely vibration-tested. They were operated continuously for over 4 hr on a vibration table at 1/32-in. amplitude and 1800 rpm, with a 1000-ft magazine attached and without a failure of any kind. They were also subjected to continuous operation under constant acceleration on a rotating table. This operation was successful up to 11.0 g about all axes except + forward, which was satisfactory up to 4.0 g. The cameras have also been operated in the cold room down to —70 F while in continuous 30-sec interval operation, but at this temperature the camera cannot be stopped for very long or it will not restart.

The lighting system for the flight recorder is sensitive to low temperatures, and if it is desired to go below about —5 F some lighting other than fluorescent tubes must be used or means for heating them must be provided.

■ Manometers

The manometer is an instrument of fundamental accuracy. In addition, its simplicity and adaptability make it a superior instrument for flight test work in large aircraft where inverted flight or other violent maneuvers are seldom encountered. The manometers are used for engine baffle pressures, induction system pressures, nacelle pressures, heating system pressures, cabin pressures, wing drag measurements with rakes, and pressure distribution studies over such items as the fuselage and cowl flaps.

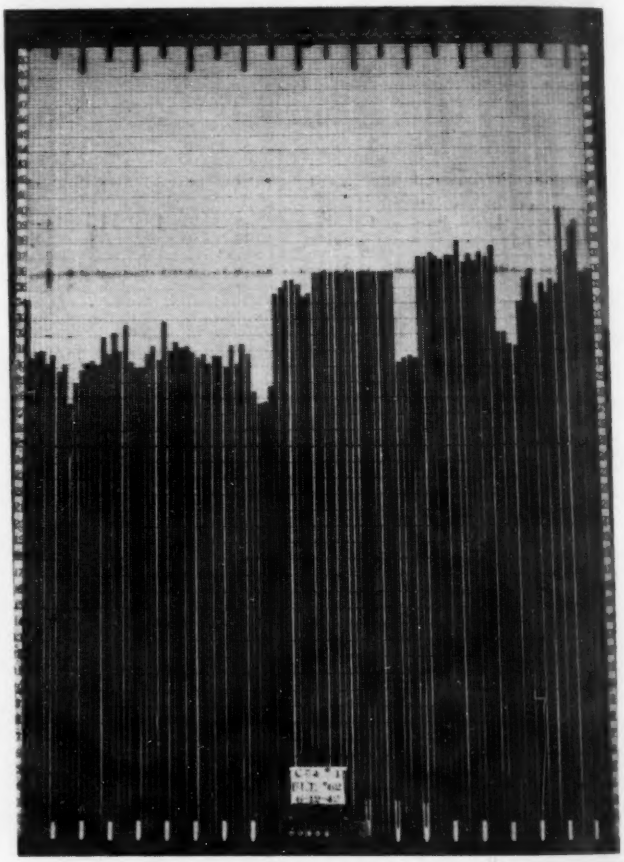
Fig. 6 shows a typical 104-tube manometer as used in our present test airplanes. The standard height is usually about 50 readable in., and the fluid used is either carbon tetrachloride (sp gr = 1.6) or alcohol (sp gr = 0.80) colored with a suitable dye such as methylene blue in alcohol or an aniline dye in the carbon tetrachloride. The tubes are made of glass of 5-mm outside diameter and 3-mm inside diameter. Each manometer usually has two or more adjustable reservoirs so that suitable reservoir level may be obtained.

Each manometer tube is separately connected to its reservoir. This method permits uninterrupted operation of the remainder of the tubes in the panel in case unexpected high pressures develop that cause the fluid in any tube or tubes to go under the base of the "U." Each manometer tube also includes a damping tube approximately 6 in. long with a 0.055-in. orifice. This type of orifice is preferred to a short one of small diameter, due to possible stoppage.

■ Automatic Temperature Recording System

The automatic temperature recording system used by the Douglas Co. came into being in April, 1941. It consists of a null-type self-balancing electronic potentiometer, an automatic switching unit, and a master control panel. We believe that the successful development of this system represents one of the greatest single improvements in flight-test equipment in recent years.

Prior to the advent of this completely automatic system, it was necessary for the flight engineer to use a manually balanced laboratory-type potentiometer and manual switching from one thermocouple to another or a slow self-balancing potentiometer available for a limited number of thermocouples only.



■ Fig. 6 – Typical 104-tube manometer panel picture

The potentiometer itself was constructed by the Brown Instrument Co. at the suggestion of the Douglas Co., and the automatic switching panel and control panel were developed by the Douglas Co. to complete the system.

The potentiometer consists of a standard Brown Recording instrument to which has been added an electronic control circuit consisting of an amplifier and other associated equipment. It operates in such a manner that the printing head is automatically brought to the correct balance position, at which time the printing bar is brought into contact with a paper roll, making a permanent record. Simultaneously with the printing, a mechanically interlocked switch provides an electrical impulse that switches the recorder to the next thermocouple circuit through the switching panel.

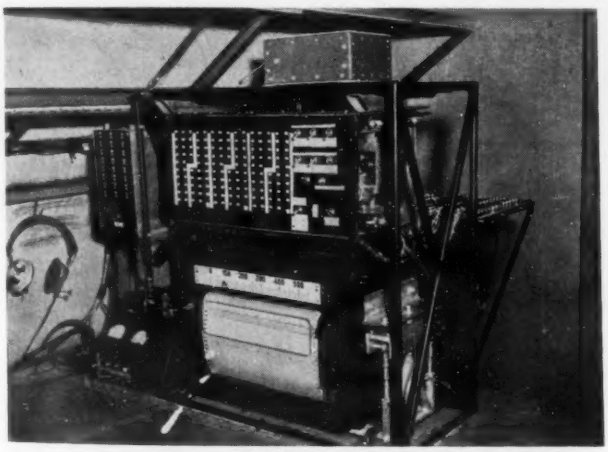
On a recent test, this potentiometer averaged 1.1 sec per thermocouple for 168 temperatures of all kinds. The

switches were grouped to keep the number of swings from high temperatures to low temperatures to a minimum.

The automatic switching unit consists of 25-point stepping relay switches, which were originally employed by automatic telephone exchanges. It is used to select the thermocouple for measurement in progressive order, 24 couples to each unit. The control panel governs the selection of switching units as desired. The whole system in turn is controlled by an impulse obtained from the potentiometer, which is supplied by the instrument at the instant it has stabilized and recorded the temperature of the point being measured.

The control panel is arranged in such a way that any desired switching unit can be continuously recycled, or any desired thermocouple can be continuously cycled, or any desired group of switching units can be recycled. These features are very useful in flight testing, since we take care to arrange the thermocouple hook-up so that all related couples are closely grouped, and thus not all the switching units need be in operation all the time.

Fig. 7 shows one of the automatic temperature recording systems as installed in the Model C-54. The switching units and control box are located above the potentiometer, which is shock mounted inside the steel framework. This particular potentiometer recorded temperatures from seven switch banks, making a total of 168 thermocouples. Only 24 of the 25 switches on the stepping relays can be used for thermocouples since the 25th switch is used to sequence to the next bank.

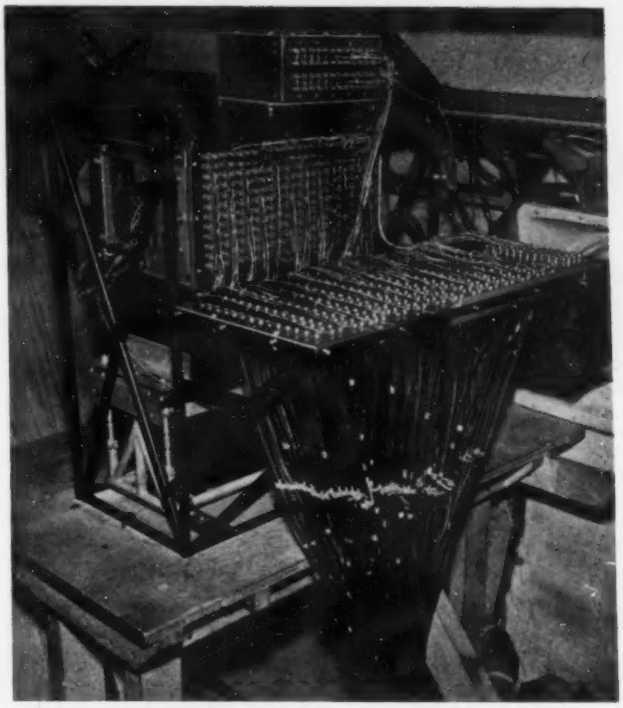


■ Fig. 7 – Main automatic potentiometer installation and the seven switching units – in the cabin

Fig. 8 is a rear view of the potentiometer shown in Fig. 7. This photograph shows the thermocouple connector panels. In this picture, the protective leather covering has been removed. The thermocouples shown are all iron-constantan.

The potentiometer has been thoroughly tested in hundreds of hours of flight and also in a wide variety of laboratory uses. It will operate continuously without any loss of speed or accuracy in temperatures down to approximately 0 F, but if it is stopped and cooled down to 0 F, it will not begin to operate again until reaching approximately 35 F, and regains its accuracy at about 45 F. The accuracy of the potentiometer is always within 0.1 of 1% of the scale range when running under normal conditions.

The error in the switching units may be from 0 to 3 F, attributable mostly to the switch contact points. In order to keep the switches in good condition and decrease the required servicing, they are constructed in hermetically sealed cases and a desiccator is employed to exclude moisture.



■ Fig. 8 – Thermocouple lead wire connecting panels in front of the switching units – with the cover off

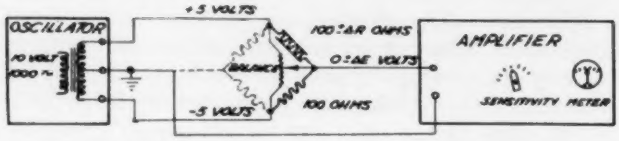
■ Strain Gages

The adaptation of electrical strain-gage circuits to flight-test usage has proceeded rapidly in the last three years until now these gages are widely employed, and we are discovering new applications each day.

The strain-measuring equipment used is essentially a resistance-change indicator of sufficient sensitivity to record the minute changes in resistance of a fine wire cemented down to the part under strain.

The "copel" wire used has a fairly high resistivity and a very low temperature coefficient, and with the methyl methacrylate cement used, exhibits a strain sensitivity of $\frac{\Delta R}{R} / \frac{\Delta L}{L} = 2.0$. The change in resistance is detected by connecting the gage in a Wheatstone bridge potentiometer circuit energized by a constant 1000-cps carrier of 10 v, and amplifying the bridge unbalance voltage with a very stable vacuum tube amplifier to sufficient strength that rugged meters can be deflected.

It will be noticed that the bridge circuit has one resistance gage and one opposing resistance of the same value, which must be assumed constant and theoretically equal in value to the unstressed gage resistance. For reasons of



economy, symmetry of wiring, temperature compensation, and other conveniences to be mentioned, it is customary with our equipment to use another strain resistance gage for this balancing unit. It is installed in the vicinity of the strain measuring gage on a small unstressed piece of the same kind of material as that under test. It is found practicable to manufacture gages with resistance tolerances in the neighborhood of 0.1%, our standard gage being a 100-ohm grid of 2-mil copel wire 16 in. long but wound into an area 1-in. long by ⅜-in. wide on rice paper.

The final balancing adjustment is accomplished by a high-resistance potentiometer shunted across the bridge circuit so the meter pointer can be adjusted at will to any convenient point on the scale corresponding to a zero-strain condition of the gage. It is an established convention always to balance our controls to a position clockwise of the zero-deflection point, usually to the point where zero stress corresponds to mid-scale reading of the meter. Then standard wiring connections will produce an increasing meter deflection for tension strains and a decreasing meter deflection (left of mid-scale) for compression strains. Obviously, different adjustments of amplifier sensitivity require different degrees of unbalance of the balancing potentiometer to maintain mid-scale meter readings for zero-strain position.

In addition to a resistance balance of bridge circuits, the magnitude of quantities involved makes it necessary to obtain a capacity balance of wiring, and so forth, in order to eliminate a confusing voltage, which would be caused by out-of-phase currents resulting from these capacitances. This is accomplished by small radio-type trimmer condensers, which are adjusted at the time of installation and need not be touched again.

Further data on the subject of strain-gage circuits may be obtained in another article[2].

Fortunately, the flexibility of the electrical circuits of these gages allows them to be adapted easily to a variety of uses in flight testing.

Since meter deflections are dependent only upon the percentage change in resistance of a bridge leg, it obviously makes no difference how many gages are used in series, as long as balance is maintained by using the same number in each leg. This fact makes it possible to apply several gages to a non-uniformly stressed part to obtain the average strain throughout the part and determine net loads; for example, net tension in a section of spar cap can be measured, in spite of simultaneous bending, by a gage on each side of the spar cap. Wrinkling effects in skin can be eliminated or net landing load in a tubular strut averaged, and so forth. Conversely, the net linear strain in a part can be canceled out and a reading obtained of difference in strain between two gage locations by actually using the dummy gage in the opposite leg of the Wheatstone bridge circuit as an active gage. This method allows cantilever beams to be calibrated and used for measuring such forces and torques as control-surface-hinge moments. In the latter example, control horns can be equipped with gages to measure the torque applied by each horn, and the two horns wired with opposite gages in series so that hinge-moment loads will be additive while cable-rig loads will cancel, resulting in a measuring unit independent of temperature, rig load, or control-surface position.

The cantilever-beam circuit may also be used for indicating positions and for recording amplitudes of vibration, and, of course, the circuits mentioned above can be used to indicate sums of differences of those motions, and so forth.

In addition to the uses mentioned above, the strain gages are regularly used for:

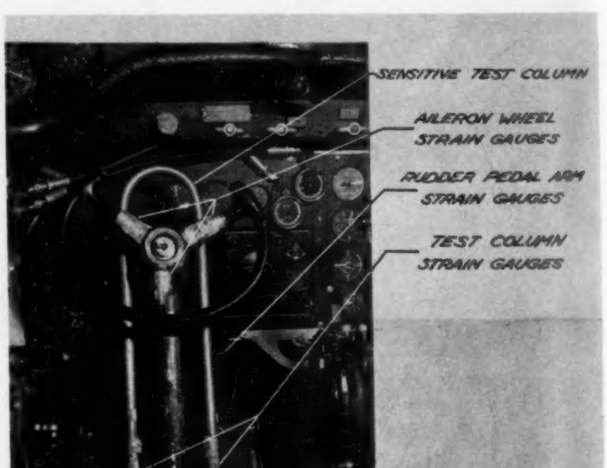
(a) obtaining pilot forces on the aileron wheel, elevator column, and rudder pedals.

(b) oleo strut material stresses.

(c) axle bending stresses to indicate when the ship leaves the ground on take-off and when it contacts the ground on landing.

(d) high-frequency-vibration investigations, where the frequency exceeds about 140 cps, the usable limit of the variable reluctance accelerometers.

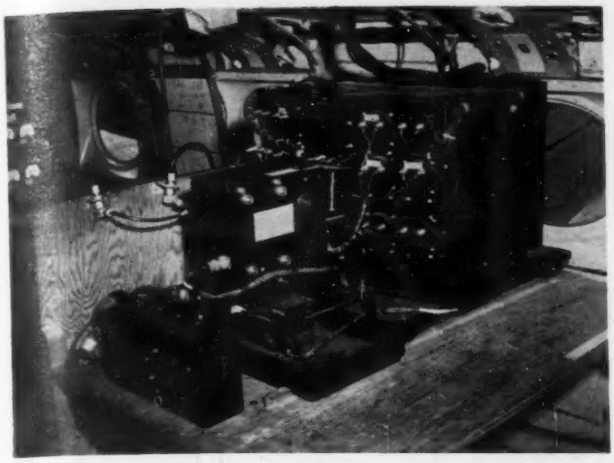
An example of the use of strain gages for airplane control system tests is shown in Fig. 9. The aileron wheel control system, rudder pedal, and sensitive elevator column strain-gage installations may be seen. Thus a complete and instantaneous record of all the forces that the pilot must apply can be obtained for any kind of flight maneuver.



▪ Fig. 9 – View of the pilot's controls showing control-force strain gages on the test column, aileron wheel, and rudder-pedal arms

Where vibration tests are conducted, the strain-gage-voltage fluctuations are recorded on film in a 12-channel mirror-type oscillograph. Where the fluctuations are comparatively steady, such as in control-system tests, we have found it more convenient to use a flight recorder camera and panel to photograph the 12 meters directly. Such an installation is shown in Fig. 10.

[2] See *Instruments*, Vol. 15, No. 4, April, 1942, pp. 112-114, 136-137: "Characteristics and Aircraft Applications of Wire Resistance Strain Gages," by A. V. deForest.



■ Fig. 10 – Strain-gage flight recorder in the rear of the cabin

This figure also illustrates what we call our "small" flight recorder panel. In this type of recorder, the panel is photographed in a mirror. The panel size is approximately 16 x 20 in. and mounts about 18 standard instruments.

Our experiences in the development of the strain-gage technique as applied to the stresses in airplane structures in flight are amply covered by Strang[8].

■ Hydraulic System Equipment

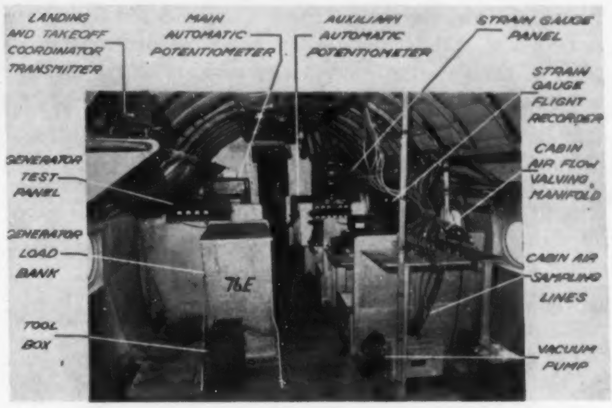
Hydraulic system performance is extremely important in our airplanes because many important components are hydraulically actuated, including landing gears, wing flaps, cowl flaps, brakes, nose wheel, steering cylinders, gun turrets, and often engine controls, flying surfaces, and landing lights. Here our flight recorder system provides the means by which information can be obtained mechanically that it is impossible to obtain by manual recording.

For this application, the flight recorder instrument panel consists of hydraulic strut-position indicators, system pressure gages, and a system temperature gage. Then by "teeing" into the desired operating pressure lines close to the unit to eliminate the line losses and photographing the panel at 10 to 20 frames per sec, we can obtain accurate load-stroke curves for every hydraulic strut in the airplane during its actual operation. By comparing such tests with the results of ground tests, the effect of the flight loads can be measured and the design criteria revised to accommodate them.

■ Electrical System Equipment

The electrical system is another important element of any large airplane. Where auxiliary internal-combustion engines are used to provide the power for the electrical energy, these engines must be flight tested for cooling, power output, regulation, and proper governing just like the main powerplants. The test equipment used is also very similar to that used for the main powerplants.

For a typical battery and generator system, it is necessary to determine that the generator cooling is satisfactory and that the cut-off switches and voltage regulators are operating properly. The flight testing equipment for either type of system consists of the necessary thermocouples, ammeters, voltmeters, switches, and some means of measuring the electrical load. The load is usually absorbed by a series of heavy cast-iron resistors wired up in such a way that the load can be changed in 0.5-kw increments up to about 120% of the expected output. Such a load bank is shown in Fig. 11 with the letters 76E on it and surrounded by asbestos to protect the structure from the heat generated. (76E only indicates the department that constructed it.)

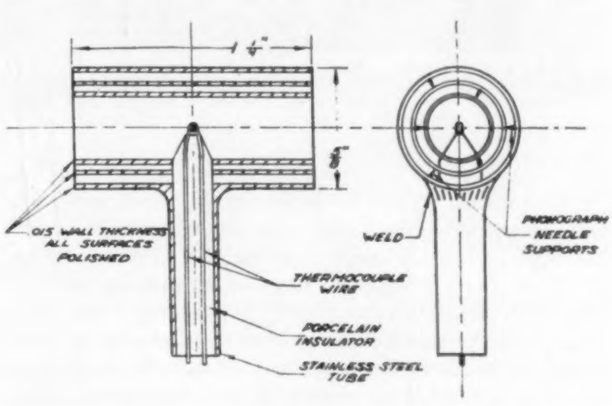


■ Fig. 11 – View of the main cabin from the rear showing the installation of test equipment prior to flight No. 15

■ Heating and Ventilating Equipment

The test equipment required for heating and ventilating tests consists of manometer panels to survey static and impact pressures at various locations throughout the ship, velometers to measure airflow through the heating and ventilating system, and thermocouples to measure the temperatures throughout the airplane. Where internal-combustion heaters are employed, fuel-air analyzers and manifold pressure gages may be necessary.

Where the air temperature inside of heated air ducts is measured, we have found that the heat radiation from the thermocouple to the cold walls is too large to be neglected. For this reason, we have developed a triple-shield unit for the iron-constantan thermocouples as shown in the sketch below.



[8] See SAE Transactions, Vol. 50, No. 8, August 1942, pp. 346-357: "Progress in Structural Design through Strain-Gage Technique," by C. R. Strang.

■ Flight-Test-Coordination Technique

The coordination of all of the above types of flight-test equipment, and many others not mentioned, is the most important single aspect of automatic recordings. This problem is solved simply by establishing a master counter and distributing other identically reading counters throughout the ship.

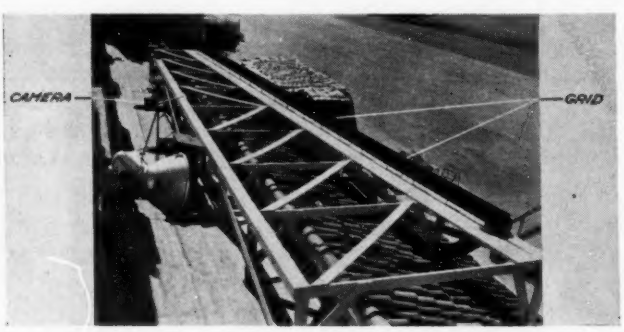
The master counter is the counter in the main flight recorder panel. This main flight recorder counter is then duplicated in the other flight recorders, at the flight engineer's panel, on the automatic temperature recorder, in the strain-gage oscillograph, at the heating and ventilating test station, and at the generator test panel. Thus every single piece of the data recorded at any position in the airplane can be correlated exactly to every other.

In addition to the counters for correlation of data, an interphone microphone and headset are installed for each member of the crew, and all conversations concerning the condition of the test equipment, airplane, or test are cleared through the chief flight engineer.

■ Take-Off and Landing Equipment

The equipment used by the Douglas Co. for measuring the take-off and landing path of a test airplane has many features that we think are noteworthy. It is a photographic method, which consists of a fixed grid permanently located on the roof of the company's El Segundo hangar. This permanent feature makes it usable on a moment's notice and keeps the equipment off the field out of the way of aircraft traffic.

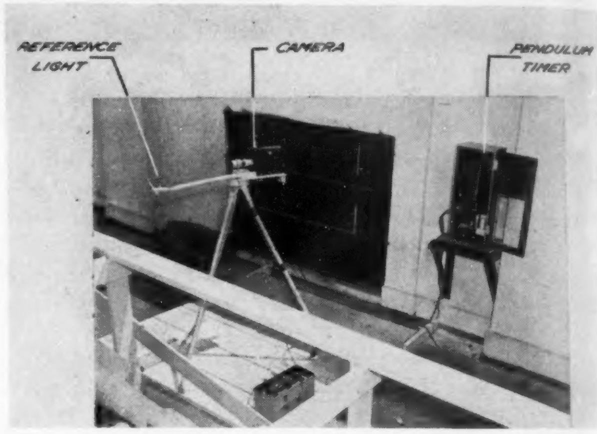
A general view of the installation is shown in Fig. 12. The grid is approximately 48 ft long and the camera is placed about 10.5 ft behind the grid. In this particular case the grid was located approximately 1300 ft from the runway, providing about 6500 ft of runway coverage. The



■ Fig. 12 – Take-off and landing grid on El Segundo hangar roof

grid was surveyed and lined up parallel to the centerline of the runway. Thus by geometry the exact position of the airplane along the runway can be determined provided the airplane is flying down a line parallel to the center line of the runway. Since the airplane in practice does not always fly down the middle of the runway, the actual position of the ship with respect to the centerline is determined by another camera located at one end of the field on the centerline of the runway or by a field observer who can closely approximate the ship's path on the runway by observation. The data can then be corrected for this variation, which is usually small.

The timing impulse for the camera is obtained from a pendulum timer shown in Fig. 13. The pendulum is located inside the box mounted on the hangar wall. Each swing of the pendulum was adjusted until the interval was 0.500 sec ± 0.002 sec, or in some cases the actual time interval over a several-minute period was determined and used. This method of exposing the pictures is considered to reduce the timing error to a negligible value, and the consistency of the actual flight-test results bears out this conclusion.



■ Fig. 13 – Take-off and landing grid camera with pendulum timer

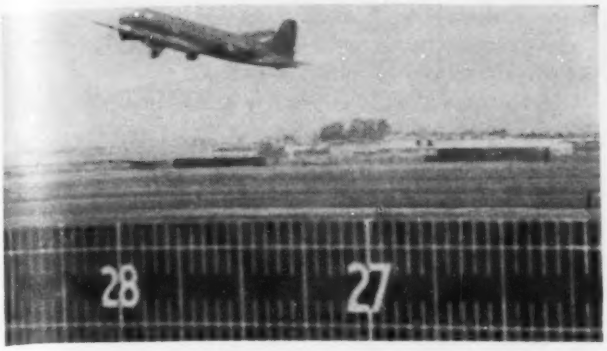
In order to establish a close relationship between the data accumulated from the grid outside the airplane and the several sources inside the airplane, a take-off and landing coordinating system was devised to turn on a light in the main flight recorder panel, simultaneously with a light on the grid camera. The main flight recorder panel light is shown in Fig. 4, and the grid camera light is shown in Fig. 13.

The control for the coordinator is located at the chief flight engineer's station, and is the oscillator control box shown in Fig. 5 below the flight recorder controls.

The operation is such that when the control button is pressed a tone lasting 0.75 sec is sent out over the ship's radio transmitter on any normal frequency (4495 kc was used). The tone is timed for 0.75 sec by an electronic time delay discharge. While the tone is being broadcast, the light in the flight recorder instrument panel goes on, and a radio receiver located at the grid camera picks up the signal at the same time. The signal received then passes through a selective amplifier, and after being amplified closes a relay, which closes the circuit to the light in the field of the grid camera. Voice modulation or static will not pass through the amplifier and trip the relay since the exact proper steady tone is required.

It will be seen that with exposures made at 0.5-sec intervals, the correlation will then be within an 0.5 sec period. The correlation could be improved somewhat, if desired, by more rapid exposures but 0.5 sec is usually a sufficiently short period.

A typical example of the type of picture obtained using



▪ Fig. 14 – Example of a picture taken with the Douglas take-off grid camera

the take-off and landing path measuring equipment described above is shown in Fig. 14.

▪ Weight and Cost of Equipment

The reason why the title of this paper includes the phrase "for large aircraft" will immediately become clear as we point out the magnitude of flight-test-equipment installations.

When the experimental airplane is nearing the final stages of construction, a flight-test interior is designed and installed. The purpose of this interior is to protect the airplane from excessive wear, damage from spillage, or dropping of heavy units. The special interior is also required to provide solid anchorage for every piece of equipment and ballast that might come loose in maneuvers. The crew must also, of course, be provided with suitable safety belts.

Heavy flooring, usually about ¾-in. thick with convenient tie-down angles, is the first to go into place. The rugged wooden tables with convenient writing space are next installed. Heavy ballast racks and tie-down covers are located in each cargo compartment and throughout the ship in sufficient quantity to ensure complete coverage through the desired range of center-of-gravity positions. Examination of Fig. 11 will reveal the type of interior that we consider suitable for flight testing.

A general idea of the amount of test equipment carried in a recent experimental airplane is given by the following actual weight data taken from a typical flight weight-and-balance sheet on the Model C-54.

Item	Weight
Test Instruments Includes also: 2 automatic potentiometers, 2 oscillographs, camera test instrument panels, generator test panel, manometer panels, strain-gage panel, trailing bomb, heat and vent panel and controls, and so forth.	2101 lb
Interior Furnishings Includes: wood flooring, asbestos sheet, ballast boxes and hold-down covers, seats, safety belts, tables, floor angles, bracing, and so forth.	4139 lb
Wiring and Tubing Includes: 12,000 ft of thermocouple wire, 4000 ft of strain-gage wire, 6000 ft of general wiring, 8000 ft of copper tubing, 1000 ft of steel hydraulic tubing, 2000 ft of electrical cable, and so on.	2395 lb
Electrical and Batteries Includes: 11 test batteries, generator load bank, lights, junction boxes, fuse panels, inverters, and so forth.	2165 lb
Flight-Test-Equipment Weight Total	10,800 lb

Every piece of equipment included above except for interior furnishings was itemized. A cost was established based on either its purchase price or, where that was not available, its listed price at the time, and the total cost came out to be approximately $38,600. This value, of course, does not include the most expensive item of all, the labor required for the installation.

▪ The Future

Although we believe we have accomplished much toward improving the accuracy of flight-test measurements and increasing the scope of all flight tests by developing mechanical recording devices, only a beginning has been made. The pathways toward further development are clearly defined in at least three directions:

1. Greater instrument accuracy must be provided without exceeding the limits of practicality.

2. The durability and reliability of the measuring machines must be extended to cover a greater range of test conditions.

3. The recording speeds must be increased without the loss of accuracy, durability, or reliability.

The science of flight testing is rapidly improving, and we can hope that as its potentialities are further grasped and understood more thought will be brought to bear on this vital subject. It is logical to suppose that the aeronautical art will advance with every improvement in the art of flight testing.

▪ Acknowledgment

We wish to acknowledge with thanks the work of the entire Research and Development Group under Richard Goldstein and Elmer Wheaton. In addition, especial appreciation is due Ralph Ostergren, Morton Moore, and Donald Barth, whose ability and help have greatly advanced the development of flight-test equipment.

The Why of Shell Metallurgy Specifications

THE "why's" of our shell metallurgy specifications are primarily the military and ballistic requirements that the shell must meet.

Military projectiles can be divided into two general classes:

1. Those that produce their effect at the target by virtue of their remaining kinetic energy, and

2. Those that produce their effect at the target principally by means of the contents of the cavities in their hollow bodies.

The only important members of the first class are the armor-piercing projectiles. These are procured under performance specifications, and hence almost no metallurgical requirements are specified.

Projectiles of the second class are usually referred to as shell. They may be divided into two general groups: special purpose shell (such as illuminating shell and smoke shell) and high-explosive shell. Let us immediately turn our attention to the high-explosive shell as being the more interesting of the two.

The most important, although not in all cases the only function of the body of a high-explosive shell, is to serve as a container to carry its contents as far as possible and with maximum accuracy and safety, to the target which is being attacked.

The shell bodies themselves are not particularly dangerous to manufacture, handle, transport, or fire. It is the active agents which they carry in their cavities that provide the element of danger. However, while the loaded shell is actually being fired from the gun, the shell body must assume a large part of the responsibility for the safety of that operation. During that short interval of time, the shell body is subjected to the forces produced by its acceleration, the pressure of the powder gases on its base and side wall behind the obturating band, and if it is being fired from a rifled gun, the additional forces produced by the pressure of the walls of the gun on the rotating band. It must withstand these forces without distorting sufficiently to initiate the action of the charge it carries, and in addition must prevent any of the powder gases from gaining access into its cavity.

Types of Weapons

The magnitude of the forces that a shell will have to withstand during firing is determined largely by the type, caliber, and power of the gun from which it is fired. In this respect, the guns used in our services can in general also be divided into two types: the smooth-bore weapons and the rifled weapons. The smooth-bore weapons are relatively low in power for their caliber. In them the maximum pressure developed by the powder gases is relatively low, and as the shell that are fired from them are not required to have rotating bands, they are not subjected to the forces produced by the action of the wall of the gun on these bands. The rifled weapons, on the other hand, have for their caliber relatively higher power, and operate at relatively high maximum gas pressures. The shells that are fired from them must have rotating bands, and hence are subjected to the forces produced by the action of the walls of the gun on these bands. Thus, the shell that are fired from rifled weapons are, in general, subjected to greater forces during firing than are those that are fired from smooth-bore weapons.

High-explosive shells carry in their cavities bursting charges of explosives which are comparatively sensitive to shock, and their action is likely to be initiated by a comparatively small distortion of the shell body or by leakage of the propellant gases into the cavity. Furthermore, if the initiation of such a bursting charge takes place prematurely, the damage to nearby personnel and materiel is likely to be very serious. In one careful study, the conclusion was reached that if any model of high-explosive shell body is the cause of more than one bore premature in 1,250,000 rounds, that model must be considered unsatisfactory from the standpoint of safety.

High-explosive shell are nearly always used to produce either demolition effect or fragmentation effect. To produce demolition effect, the shells are provided with delay action fuses, which allow them to penetrate into targets such as buildings and dugouts, before they cause the bursting charges to explode. The extent of the effect they will produce when so used is dependent upon the amount of explosive they carry in their bursting charge. Consequently, in shells that are designed primarily to produce demolition effect, the cavities should be as large as other considerations will permit.

Fragmentation Effect

To produce fragmentation effect, the shells are fitted with time or super-quick impact fuses, which cause the bursting charges to explode either while the shell are still in the air or immediately upon coming in contact with any solid object. The detonation of the bursting charge breaks the shell body into fragments and propels these fragments away from the point of burst with very high initial velocities. It is these rapidly moving fragments that in this case produce the effect. In shells that are designed primarily to produce fragmentation effect, the capacity of the cavity could be made somewhat smaller if only the maximum fragmentation effect were considered. However, there is another consideration, and with regard to this I quote the late General Rohne of the German Army concerning World War I:

"The fundamental difference between the high-explosive shells used with the German and French field guns, is that the German shell has thick walls and a small bursting charge, while the French shell has thin walls with a heavy bursting charge. The German shell obtains greater effect than the French shells by its fragments; the latter, however, by the tearing force and noise of its explosions, cause a marked effect on the morale of troops under fire, as if they had been targets for large-caliber guns. Troops that previously have stood up well under shrapnel fire, do not show up as well under French high-explosive fire. The effect on morale, the extent of which cannot be imagined, caused quite a surprise. The heavy bursting charge also produced considerable effect on materiel."

Thus we see that for maximum efficiency, high-explosive shell, both demolition and fragmentation, should have large cavities. Shells so designed are known as high-capacity shells.

Now the outside diameter of a shell is fixed by the caliber of the gun from which it is to be fired, and its length is limited by the requirement for stability in its flight through the air. Consequently, as the cavity is made larger, the walls must be made thinner. As this is done the unit stresses that are induced in the steel in base and side walls by the forces that act upon the shell during firing are raised.

Excerpts from the paper of the same title by Col. H. H. Zornig, Ordnance Department, U. S. Army, presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 12, 1943.

STORAGE BATTERY PERFORMANCE AT LOW TEMPERATURES

by JOHN H. LITTLE[1] and ROBERT A. DAILY[2]

THE operation of motor vehicles in subzero temperatures presents a number of problems which must be recognized and understood. It is obvious that the first general problem is that of getting the engine into operation. The ability of the engine to start is dependent upon a number of factors such as type of fuel, lubrication, condition of the engine, adequate ignition (for a gasoline engine), and cranking speed. The last two are dependent upon the battery, and it is this piece of equipment which we propose to discuss.

Inasmuch as the electric storage battery is essentially a chemical device, its activity is a function of its temperature. It will perform better when it is warm than when it is cold. Because of this characteristic, it must not be assumed that the battery cannot be used when cold, as it can be used in this condition if too much is not demanded of it, and its limitations are recognized.

The starting of an engine is the most severe duty the battery has to perform. Even at summer temperatures, the current draw from the battery is quite high, and when the engine is cold, this draw becomes increasingly greater, and at subzero temperatures will be of the order of a few hundred amperes for small engines to over a thousand amperes for diesel engines. The energy delivered by the battery on starting, expressed in watt-hours or ampere-hours, is relatively small in spite of the large current draw, as the time of draw is small. In fact, as will be shown later, this heavy current draw cannot be obtained for any great length of time.

The performance of the battery is, in reality, dependent on three factors: temperature, state of charge (specific gravity of electrolyte), and previous history. The last factor will be disregarded in this discussion, as it involves many variables which are usually unknown. It involves anything that has occurred to shorten the life of the battery, such as freezing, overheating, overcharging, buckling of the plates, loss of active material from the plates, remaining in a state of discharge for extended periods.

The battery should be kept in as high a state of charge as possible. At subzero temperatures this is important, also, to prevent freezing of the electrolyte. From Fig. 1, showing the freezing temperatures of sulfuric-acid electrolyte, it is seen that acid of 1.300 sp gr freezes at −95 F. The acid density of a fully charged battery is usually approximately 1.275 sp gr, and this will freeze at −90 F. A battery that has been allowed to drop to a sp gr of 1.220 will freeze at −30 F. As −30 F is not an unusual tem-

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

[1] Electrical engineer, Chevrolet Division, General Motors Corp.
[2] Battery engineer, Delco-Remy Division, General Motors Corp.

PLENTY of heat is the best solution to the problems encountered in using storage batteries at low temperatures, the authors say.

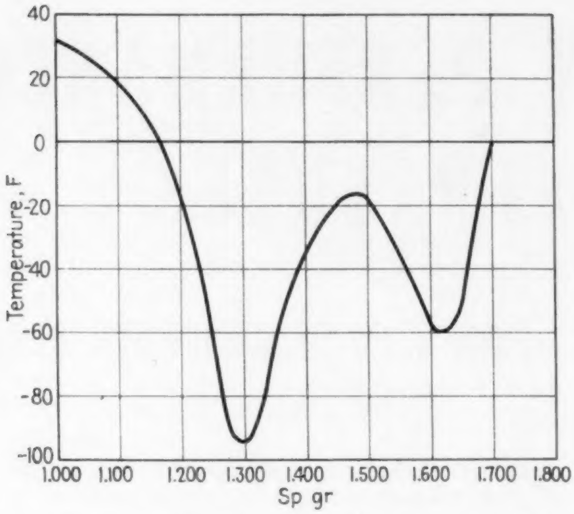
Starting at low temperatures is difficult for two reasons. A cold engine requires higher torque to start it, and the colder the battery the less the amount of current that can be drawn from it.

With a 24-v battery, an engine requires to start it at a certain temperature only one-fourth the amount of current it would require if the battery were only 6 v.

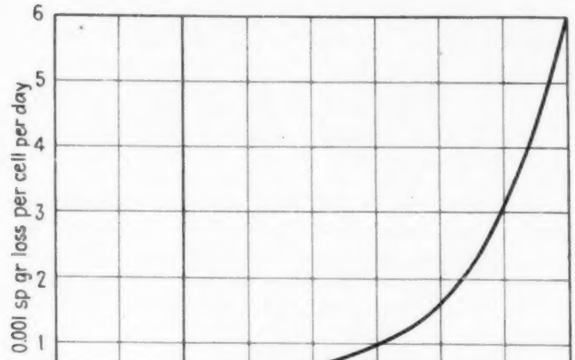
After an engine has been started, it is necessary to put back the energy taken out plus a little more because battery charging is not 100% efficient. This cannot be done at low temperatures. For efficient charging, the battery temperature should be at least 40 F.

For batteries that are used fairly frequently, it is helpful to keep them in well-insulated boxes, or in boxes that have had a piping system rigged up so that warm water can be circulated through the boxes. Another solution, satisfactory for batteries having rubber cases but not for those having composition cases, is to use a suitable heater. If a battery has been parked for days, even a well-insulated box is not enough, so that some sort of heater must be used. One type of heater in successful use is the Coleman No. 520 gasoline burner, which supplies 5000 Btu per hr. About 45 min should be allowed to heat the battery with this burner, before it can be used for starting.

THE AUTHORS: JOHN H. LITTLE graduated from the Massachusetts Institute of Technology in 1923 with a B.S. degree in electrochemical engineering. Following his graduation he became research engineer for Johns-Manville, Inc. He joined the Electrical Section of the General Motors Research Laboratory in 1925, and has been successively assistant head of the Lighting Section of General Motors Research Laboratory, assistant electrical engineer of the Chevrolet Division of General Motors, and electrical engineer of the Chevrolet Division, the position which he now holds. ROBERT A. DAILY has been with the Delco-Remy Division of General Motors Corp. for 12 years. His present position is that of storage battery engineer. Mr. Daily was graduated with a B.S. degree in chemical engineering from Purdue University.



■ Fig. 1 – Freezing points of sulfuric-acid electrolyte (adapted from Arendt)



■ Fig. 2 – Effect of temperature on rate of self-discharge

perature in some localities, the importance of keeping the battery fully charged is readily understood.

Of lesser importance at low temperatures is the self-discharge rate shown in Fig. 2. In vehicles parked for weeks at low temperatures, this need not be a cause for concern. However, if the vehicle is parked in high temperatures, it must be taken into account. A fully charged battery of 1.275 sp gr, if allowed to stand in an ambient temperature of 100 F for 25 days, would be down to 1.200 sp gr.

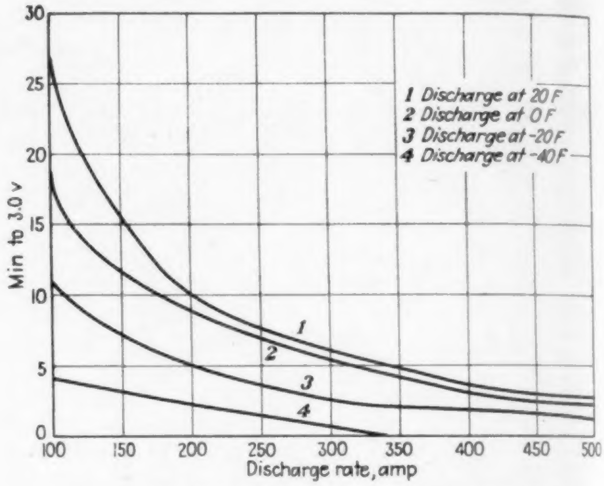
In the course of this paper, reference will be made to the following four military-type batteries:

Type	Delco Model	Voltage	Capacity at 20-Hr Rate, amp-hr
2H	11606-A	6	116
4H	19-C-1	6	150
7H	25AT	6	200
8T	168-12T	12	200

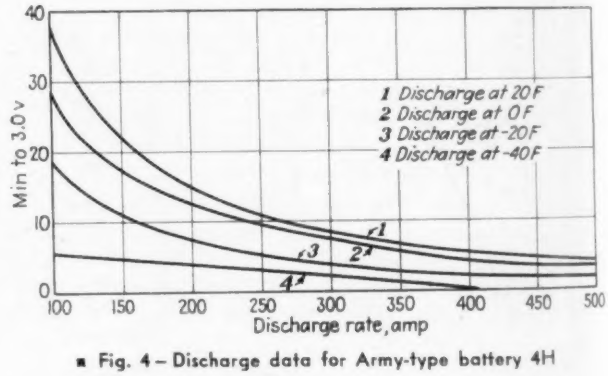
Specific data will not be given on discharge and charge curves for the 8T battery. Electrically, the 8T consists of two 7H batteries, and therefore all voltages of the 7H can be multiplied by two for the 8T.

Inasmuch as all cold cranking operations involve high-current draw from the battery, we will examine the high-rate discharge characteristics of the above batteries. For convenience, these are expressed in the number of minutes a given discharge rate can be obtained until the voltage has reached 3 on a 6-v battery. Figs. 3, 4, and 5 are such curves for the fully charged 2H, 4H, and 7H batteries respectively.

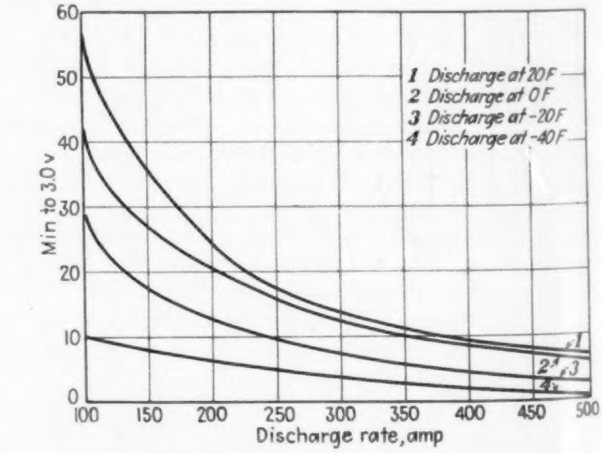
Let us consider an engine requiring 450 amp for cranking at −40 F. From Figs. 3 and 4, it is evident that neither a 2H nor a 4H battery could be used if the battery is at −40 F. However, with these batteries warmed up to −20 F, they would be usable. The 7H battery would



■ Fig. 3 – Discharge data for Army-type battery 2H



■ Fig. 4 – Discharge data for Army-type battery 4H



■ Fig. 5 – Discharge data for Army-type battery 7H

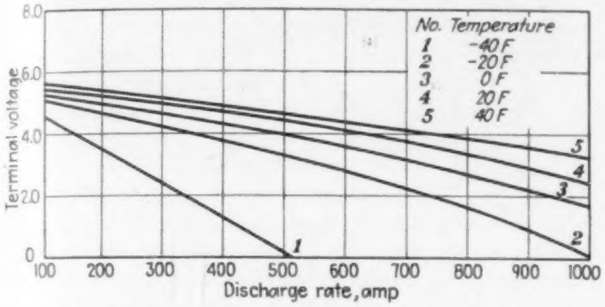
supply this current at —40 F for about one minute, but the cranking performance would be improved with the battery at —20 F.

Motor vehicles, including military types, are using 6-, 12-, and 24-v electrical systems at present. A little study of the curves will readily prove the advantage of the higher-voltage system for starting. To cite a specific case, an engine which, with a 6-v battery, would require approximately 480 amp to crank, requires only about 120 amp with a 24-v battery. In the latter case, four 7H batteries are used, and the cranking is quite readily accomplished with the batteries at —40 F. This particular case involves a gasoline engine, and ignition voltage during cranking is approximately 19 to 20, which is quite satisfactory.

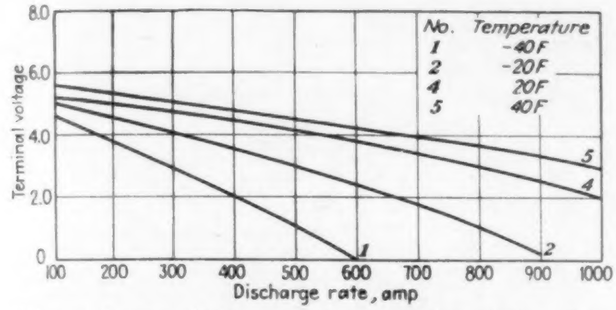
As a general statement, it is safe to say that none of the above batteries can be used for a current drain of more than 200 amp for cranking at —40 F.

Figs. 6 to 14 show the 5-sec voltages for the 2H, 4H, and 7H batteries for three states of charge at various temperatures and up to 1000-amp discharge rate. These curves, in some cases, include a comparison between wood and rubber separators. At the higher discharge rates, the rubber separators show some improvement over the wood separators.

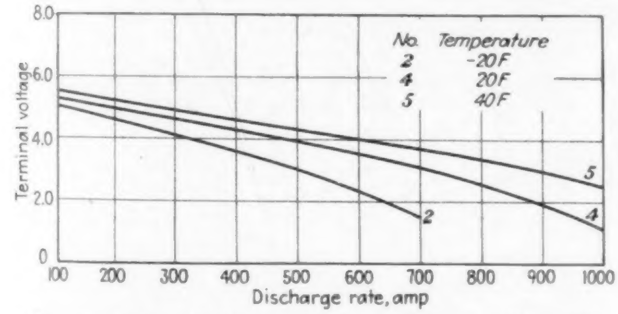
These curves are useful in coming to some conclusion



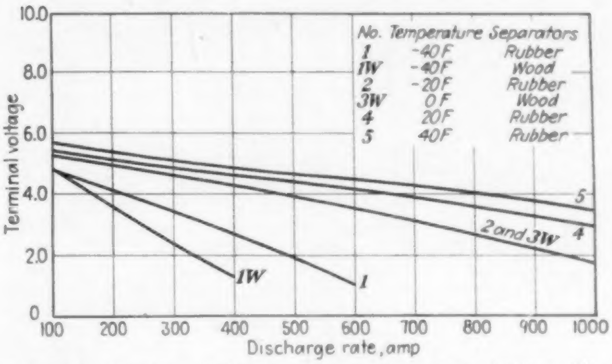
■ Fig. 6 – Five-sec discharge voltage – Army-type battery 2H – filling acid 1.275 sp gr, rates at full charge



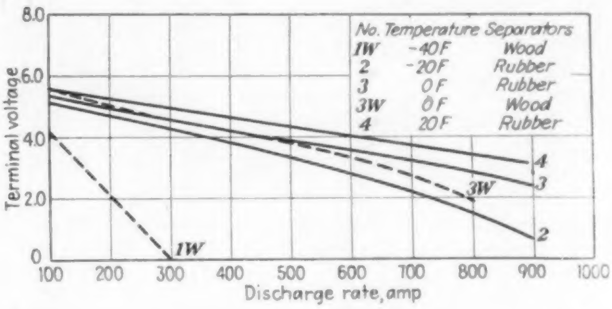
■ Fig. 7 – Five-sec discharge voltage – Army-type battery 2H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)



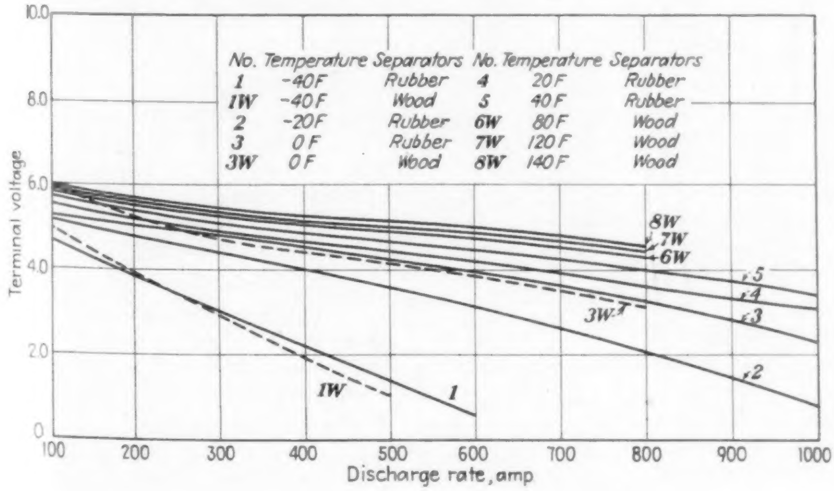
■ Fig. 8 – Five-sec discharge voltage – Army-type battery 2H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)



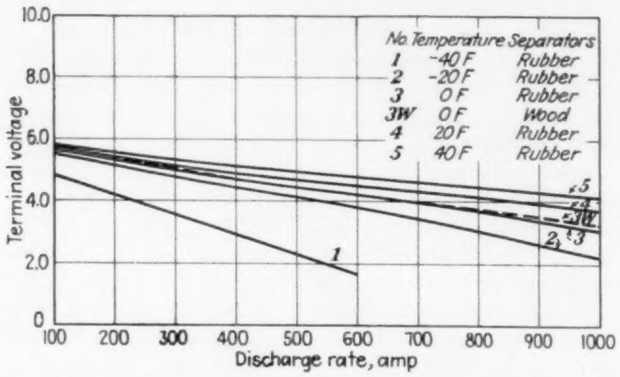
■ Fig. 10 – Five-sec discharge voltage – Army-type battery 4H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)



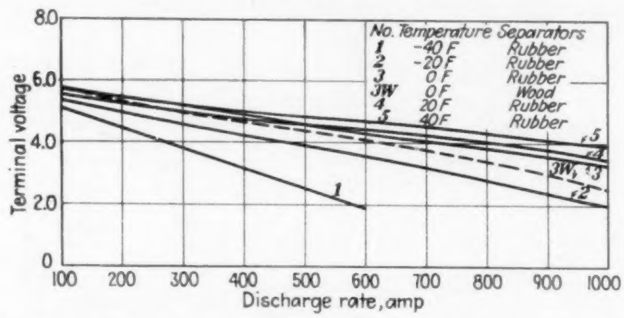
■ Fig. 11 – Five-sec discharge voltage – Army-type battery 4H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)



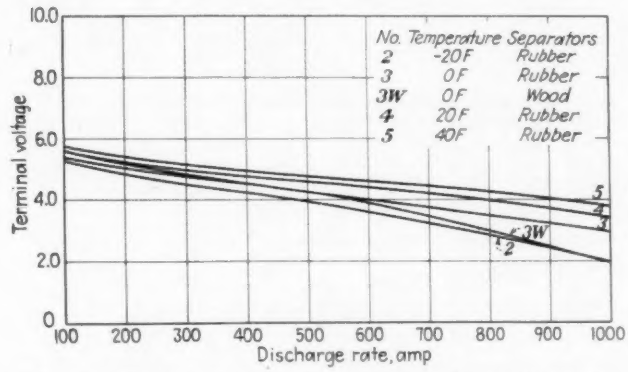
■ Fig. 9 – Five-sec discharge voltage – Army-type battery 4H – filling acid 1.275 sp gr, rates at full charge



■ Fig. 12 – Five-sec discharge voltage – Army-type battery 7H – filling acid 1.275 sp gr, rates at full charge



■ Fig. 13 – Five-sec discharge voltage – Army-type battery 7H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)

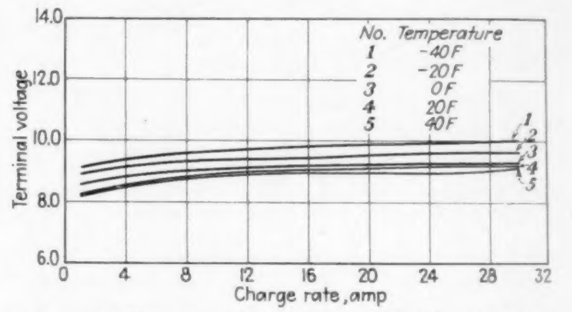


■ Fig. 14 – Five-sec discharge voltage – Army-type battery 7H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)

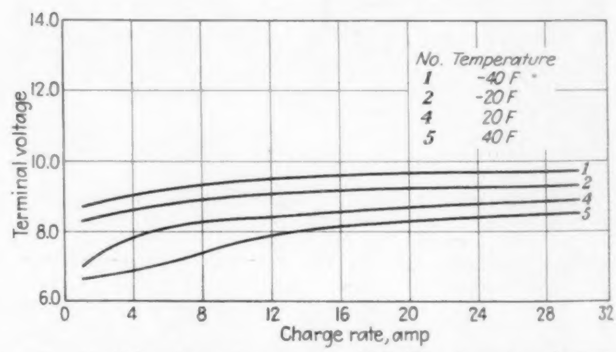
as to the battery temperature necessary to perform a given cranking operation. For example, if we decide that the cranking current is 1000 amp, then we should select a battery temperature which will give a 5-sec voltage of at least 4. From Fig. 12 for the 7H battery, we find that the battery temperature should be about 20 F. It is obvious that by paralleling batteries, lower temperatures and lower system voltages may be used. Briefly, the higher the discharge rate, the higher the battery temperature must be.

Once the engine has been started, the energy that has been removed from the battery in cranking should be replaced by charging. This cannot be accomplished with the usual generator systems, unless the battery temperature is raised. In Figs. 15 to 23 are shown the charging voltage curves (CEMF) for the 2H, 4H, and 7H batteries at various temperatures and specific gravities. In replacing the energy removed from the battery, more must be added than has been taken out. Fig. 24 shows the efficiency of the charge at various temperatures for a 4H battery; and from this it can be seen that 90% is the maximum efficiency. With a cranking current of 480 amp for 10 sec, 1 1/3 amp-hr have been removed, and 1½ amp-hr will be required to replace this energy.

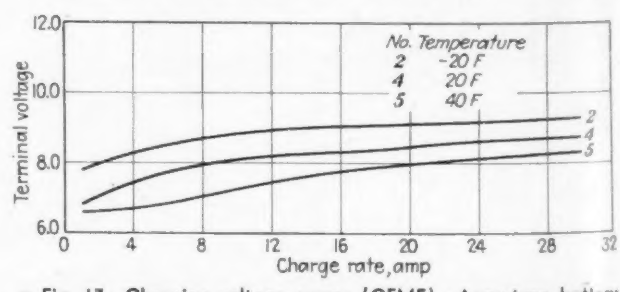
With the usual 6-v generating systems, the maximum voltage may be 7.5 or less. Reference to Fig. 16 shows that for a 2H battery at a partial charge of 1.250 sp gr, a voltage of 8.7 is necessary to put 1 amp into the battery at −40 F. The generator cannot meet these conditions. If the battery is heated to 20 F, then 2.5 amp will be put into the battery at 7.5 v. The 1.250-sp gr curve has been used for illustration, because while the energy removed by the starting is quite small compared to the total available and does not materially affect the overall charge of the battery, nevertheless, the resistance of the battery at the moment is increased. This results in the battery appearing partially discharged, as far as the charging rate is concerned. The 12-v and 24-v systems may be interpreted in the same way by dividing the maximum available generator voltages by two and four respectively. As the charging progresses, the battery will appear more nearly as a fully charged battery, so the charging rate at a given voltage will be reduced. The battery temperature will, therefore, have to be in-



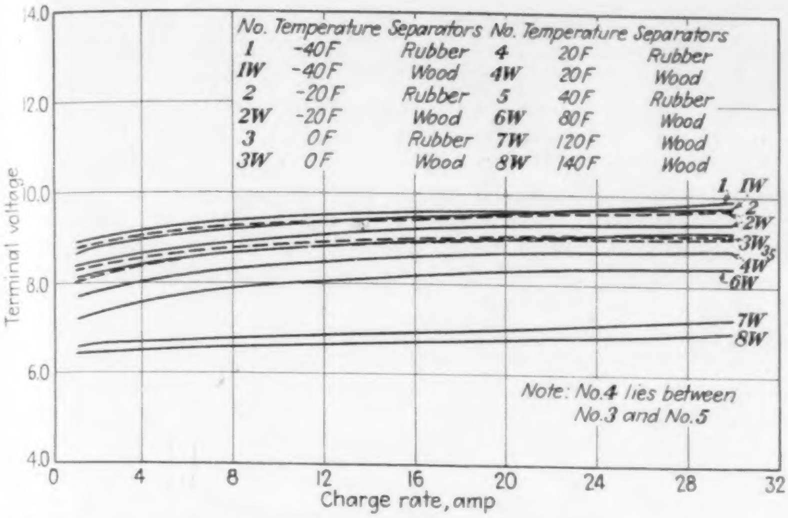
■ Fig. 15 – Charging voltage curves (CEMF) – Army-type battery 2H – filling acid 1.275 sp gr, rates at full charge



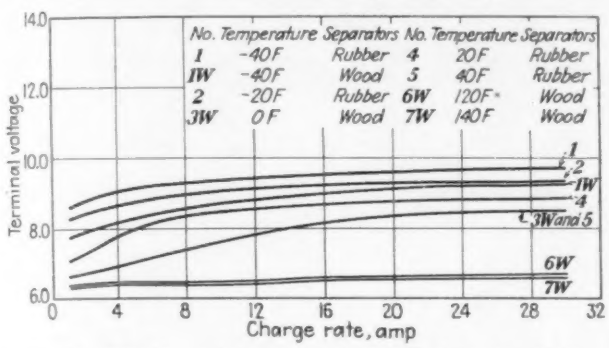
■ Fig. 16 – Charging voltage curves (CEMF) – Army-type battery 2H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)



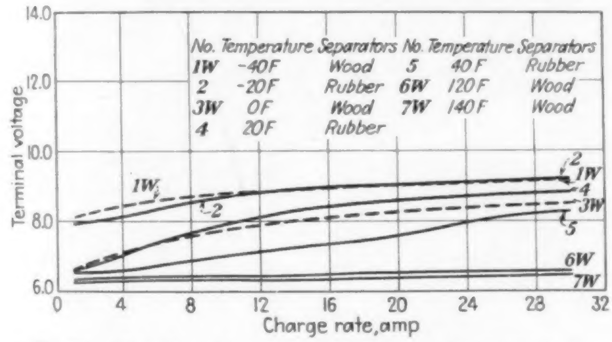
■ Fig. 17 – Charging voltage curves (CEMF) – Army-type battery 2H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)



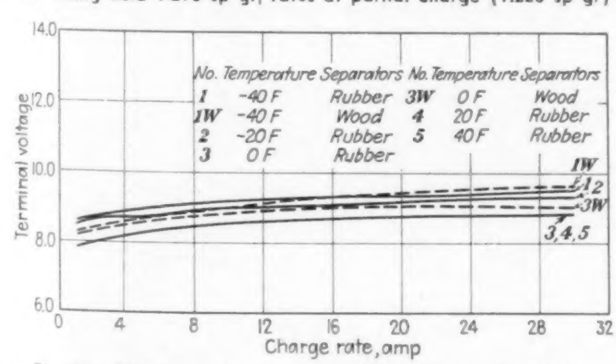
■ Fig. 18 – Charging voltage curves (CEMF) – Army-type battery 4H – filling acid 1.275 sp gr, rates at full charge



■ Fig. 19 – Charging voltage curves (CEMF) – Army-type battery 4H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)



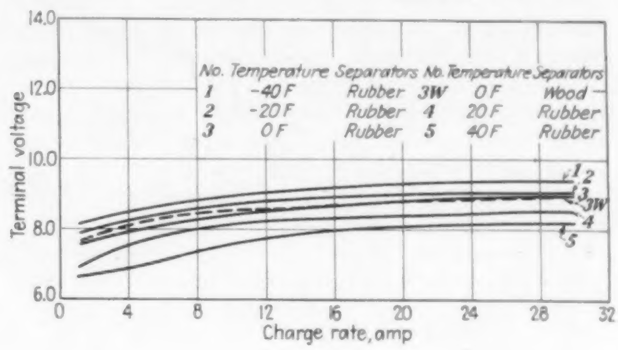
■ Fig. 20 – Charging voltage curves (CEMF) – Army-type battery 4H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)

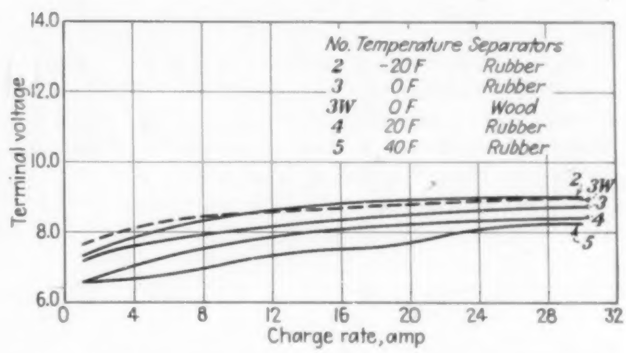


■ Fig. 21 – Charging voltage curves (CEMF) – Army-type battery 7H – filling acid 1.275 sp gr, rates at full charge

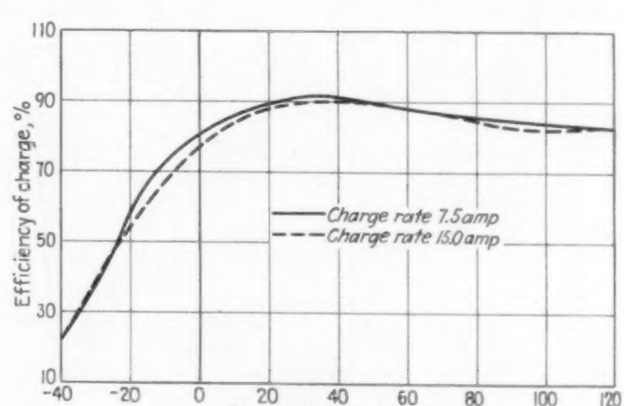
creased properly to complete the charge at the available voltage. Fig. 24 shows that for the efficient conversion of the charging current to energy stored in the battery, the

■ Fig. 22 – Charging voltage curves (CEMF) – Army-type battery 7H – filling acid 1.275 sp gr, rates at partial charge (1.250 sp gr)





■ Fig. 23 – Charging voltage curves (CEMF) – Army-type battery 7H – filling acid 1.275 sp gr, rates at partial charge (1.220 sp gr)



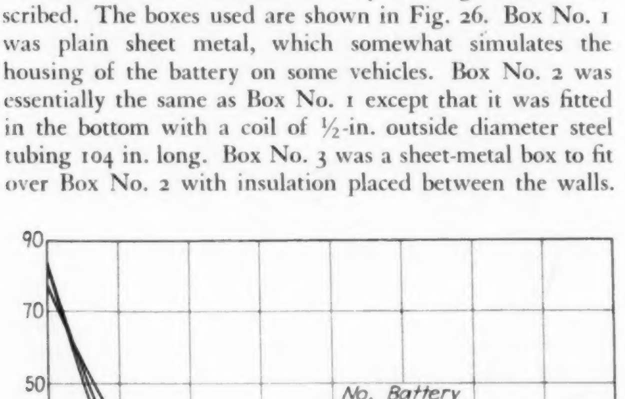
■ Fig. 24 – Efficiency of charge at various temperatures – Army-type battery 4H

temperature need not be above 20 F. The operating regulated voltage of the 6-v system is usually 7 to 7.1 v; and with a 24-v system, it is about 27.5 v. One-fourth of this is 6.87 v. This means that it will be better to warm the battery to 40 F for charging. It has been shown that the battery temperature at the time of cranking is very important, and further, that an even higher temperature is required to charge the battery.

This brings us to a consideration of the cool-down characteristics of the battery, and means for retarding the cool-down, together with means for warming the battery sufficiently for cranking purposes, and subsequently, means for further warming it while the vehicle is being used.

The heat-loss curve shown in Fig. 25 was obtained by placing batteries at room temperature (77 to 83 F) in an ambient of —40 F, and recording the temperature of the center of each of the cells. The plotted temperatures are the average of these three readings except that in the case of the 8T (168-12T) battery the average is for six cells. There is not a great difference between the three-cell batteries, but the 8T cooled down somewhat more slowly. To cool down to —20 F required 9.5, 9.5, 12.4, and 15.5 hr for the 2H, 4H, 7H, and 8T batteries respectively. If a vehicle was being operated in extremely cold weather (—40 F) and was stopped during the night for 12 hr with a battery temperature of 50 F, then the temperatures of the above batteries the next morning would be —28 F, —28 F, —24 F, and —16 F, respectively. These results are for still-air conditions and any wind would hasten the cool-down. To start the engine, the battery will require heating first or the cool-down will have to be retarded.

In the following results on cool-down and warm-up of batteries, only 4H batteries were used. The results obtained with a number of different battery housings will be described. The boxes used are shown in Fig. 26. Box No. 1 was plain sheet metal, which somewhat simulates the housing of the battery on some vehicles. Box No. 2 was essentially the same as Box No. 1 except that it was fitted in the bottom with a coil of ½-in. outside diameter steel tubing 104 in. long. Box No. 3 was a sheet-metal box to fit over Box No. 2 with insulation placed between the walls.

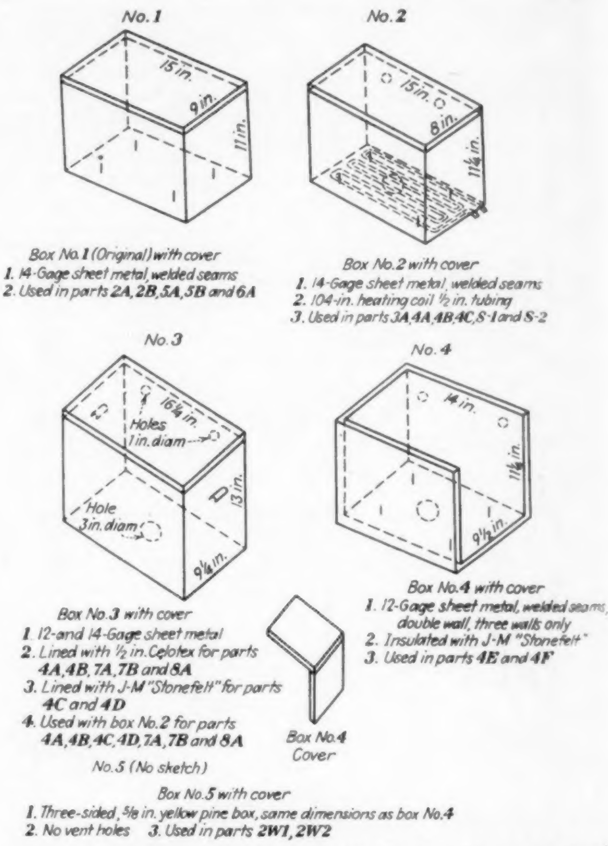


■ Fig. 25 – Heat-loss curves – ambient temperature —40 F – unprotected batteries

This insulated box had no metallic connection between the inner and outer walls. Box No. 4 consisted of an insulated metal walled box having an "L" shaped cover. The inner and outer metal boxes were fastened together with welded-on metal strips.

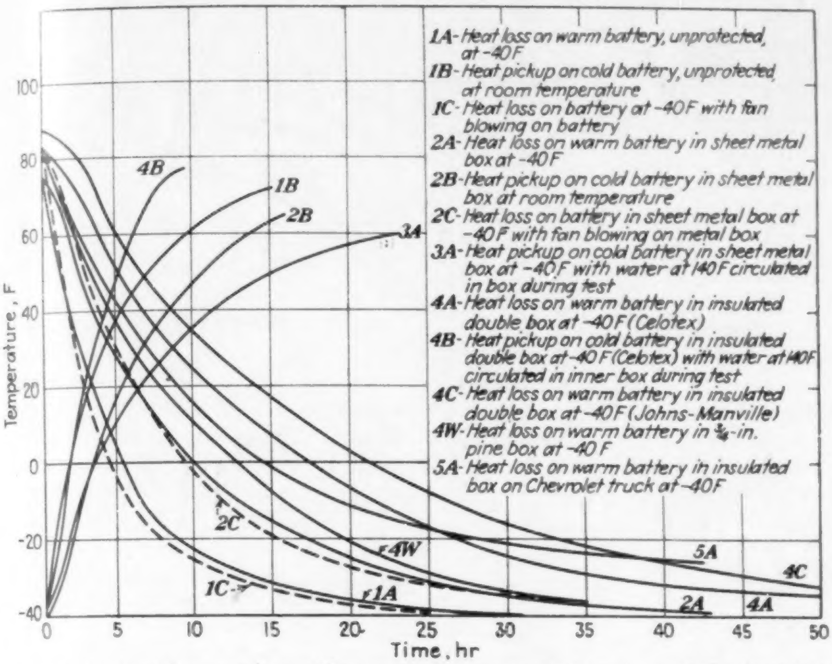
The pipe coil in the bottom of the box was used to warm the battery up by circulating 140 F water at the rate of three gal per min through it. The water circulation was started after the battery had cooled down to —40 F. The insulation used was Celotex or Johns-Manville Stonefelt (rock wool).

In comparing the curves (Fig. 27), allowance must be made for the difference in starting temperatures. Let us consider again a battery at 50 F with an ambient of —40 F

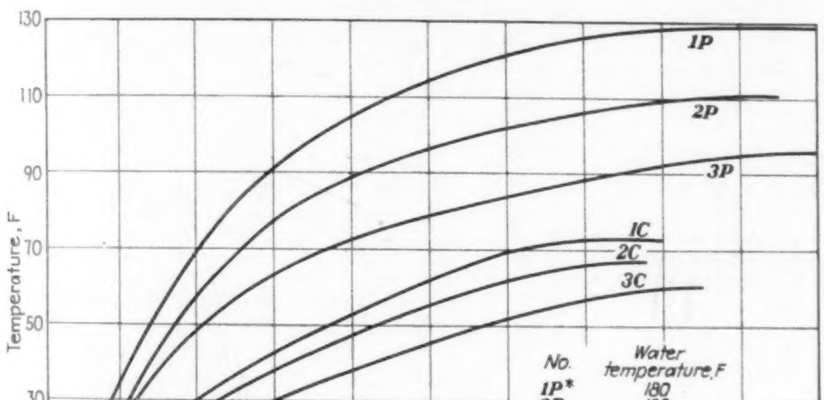


■ Fig. 26 – Boxes used in the tests (inside dimensions are given)

and note the time necessary to reach —20 F in still air. For a bare battery (curve 1A in Fig. 27), battery in a plain sheet-metal box (curve 2A), and battery in rock wool insulated double-metal box (curve 4C), the times are 8, 14.5, and 26.5 hr, respectively. Curve 5A represents the cool-down on a 4H battery in a double-metal box with rock wool insulation, having only a few rivets connecting the inner and outer boxes, with wood strips around the edges and having an "L" shaped cover. This installation was on the frame of a truck placed in a cold room. From the curve, it can be seen that the time to cool from 50 F to —20 F in an ambient of —40 F, is 24 hr. Even though the effect of considerable wind on the cool-down is not known, this type of installation appears entirely adequate for better than over-night parking. An important fact to



■ Fig. 27 – Heat-loss and heat-pickup curves – room temperature to —40 F – Army-type battery 4H

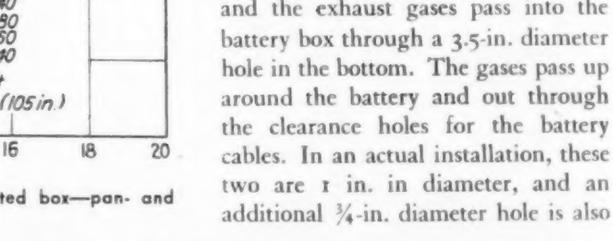


■ Fig. 28 – Heat-pickup curves – Army-type battery 4H – J. M. insulated box—pan- and coil-type units – water flow 3 gal per min

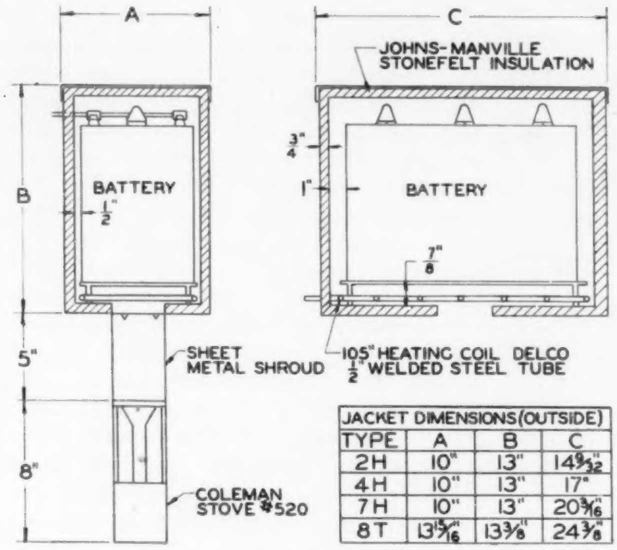
point out is that a cool-down was tried, using a plain pine box, which gave results essentially the same as Box No. 4. It is, therefore, essential that any metallic connection between the inner and outer box be kept to a bare minimum.

Curves 1B and 2B are for a bare battery and a battery in a plain metal box taken from the —40 F cold room, placed in the open room (approximately 76 F), and allowed to warm up. Curve 4B shows the warm-up of the Celotex insulated box with battery from —40 F and with the 140 F water circulating through the coil. This curve is somewhat more optimistic than data taken at a later date and shown in Fig. 28. In using the heating coil, it is placed on the inside bottom of the box and the battery rests on a tray which is supported on spacers from the bottom of the box and is not necessarily in contact with the heating coil. A second type of water circulating device was tried, consisting of an upper pan which formed the tray, and to which was stitch-welded on the underneath side another piece of sheet metal in which a passage 36 x 1 x 5/16 in. was formed. This resulted in quicker warm-up, as shown in Fig. 28, due to the battery being in more intimate contact with a hot surface. This results in what we consider too high a battery temperature during a running day, particularly with 180 F water such as might result with the higher-temperature thermostats used in the engine. It could, perhaps, be modified by raising the battery with spacers, so as not to have full contact with the tray, or perhaps less water passage space could be used. The reason for starting these curves at —10 F will be evident from the discussion which follows.

When the vehicle has been parked for days in a —40 F ambient, the battery temperature will be too low for starting, in spite of the insulated box. In this case, means must be used for bringing the battery up to a usable temperature. For this purpose, a Coleman No. 520 gasoline burner, having an output of 5000 Btu per hr, has been found to be satisfactory. This burner is shown in Fig. 29 and is fitted with a special flue which extends 5 in. above the burner when in use. It is provided with three tabs for bayoneting onto the battery hanger, and the exhaust gases pass into the battery box through a 3.5-in. diameter hole in the bottom. The gases pass up around the battery and out through the clearance holes for the battery cables. In an actual installation, these two are 1 in. in diameter, and an additional ¾-in. diameter hole is also



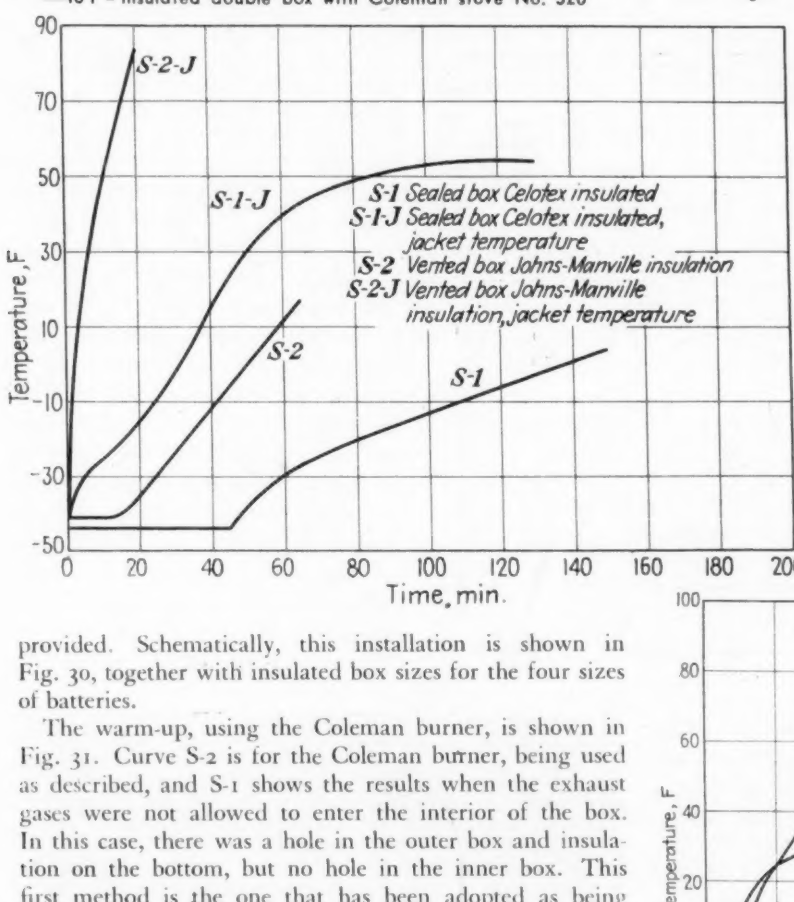
■ Fig. 29 – Coleman gasoline burner No. 520



JACKET DIMENSIONS (OUTSIDE)			
TYPE	A	B	C
2H	10"	13"	14 9/32"
4H	10"	13"	17"
7H	10"	13"	20 3/16"
8T	13 15/16"	13 3/8"	24 3/8"

■ Fig. 30 (above) – Installation of Coleman gasoline burner No. 520

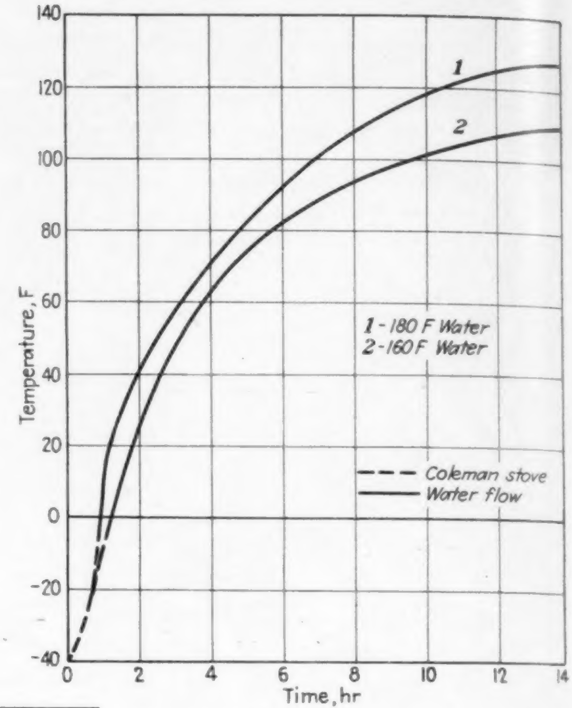
■ Fig. 31 (below) – Heat-pickup curves – room temperature from —40 F – insulated double box with Coleman stove No. 520



provided. Schematically, this installation is shown in Fig. 30, together with insulated box sizes for the four sizes of batteries.

The warm-up, using the Coleman burner, is shown in Fig. 31. Curve S-2 is for the Coleman burner, being used as described, and S-1 shows the results when the exhaust gases were not allowed to enter the interior of the box. In this case, there was a hole in the outer box and insulation on the bottom, but no hole in the inner box. This first method is the one that has been adopted as being faster. No harmful effects have been observed in the tests with this rapid warm-up, and only rubber-case batteries have been used. It would not be considered suitable with the usual composition battery cases.

From curve S-2, it is observed that the battery can be warmed up from —40 F to —10 F in approximately 40 min. In an actual installation, it could be specified that the burner be applied for at least 45 min, and this would



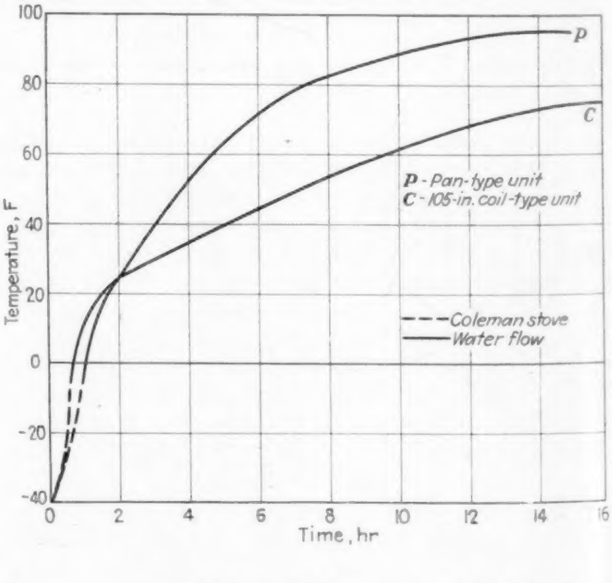
■ Fig. 32 – Heat-pickup curves – Army-type battery 4H – J. M. insulated box – Kold-hold pan unit – water flow 3 gal per min

result in an adequately warm battery for starting.

The hole in the bottom of the battery box should be covered by a slide cover or otherwise, when the burner is not being used.

Additional curves in Figs. 32 and 33 show the warm-up, using the burner from a temperature of —40 F to —10 F, and continuing with the water coil or water pan. **turn to page 164**

■ Fig. 33 – Heat-pickup curves – Army-type battery 4H – J. M. insulated box – Kold-hold pan and 105-in. coil unit – water temperature 140 F – water flow 3 gal per min



NEW MATERIALS for AIRCRAFT ENGINES

by MEL YOUNG and HERMAN H. HANINK
Wright Aeronautical Corp.

"CONSERVE strategic materials!" has been the cry dinned into the ears of the designer of wartime products; consequently, this report on new materials for aircraft engines is particularly timely. "The tremendous demands of war production," the authors say, "have now revealed our resource shortages, and in our efforts to build war and transport machines it has become necessary to take conservation measures with some elements in particular and with all materials in general."

The authors pay particular attention to new developments in the use of steels, plastics, synthetic rubber, and silver.

■ ■ ■

THE AUTHORS: MELVIN H. YOUNG (M '38), a graduate in metallurgy from the Sheffield Scientific School of Yale University, started his career in the materials laboratory of the Wright Aeronautical Corp., and changed to test engineering in 1936. After four years of engineering work devoted mainly to high output engine testing of fuels and lubricants, he returned to materials activities, and is now chief of the Materials Laboratory. HERMAN H. HANINK joined the Wright Aeronautical Corp. in 1938 after graduating from the University of Michigan with a degree in metallurgical engineering. As head of the Laboratory Heat Treat Unit he was instrumental in improving many production heat treat procedures. More recently he has worked on laboratory and production tests with aircraft alternate steels. He is now laboratory supervisor.

ACHIEVEMENTS in aircraft construction in the past 15 or 20 years appear to center about the "glamor" alloys of aluminum and magnesium, with a tendency to overshadow the important roles played by the ferrous alloys, bronzes, and any non-metallics that have forced their way into use. The tremendous demands of war production have now revealed our resource shortages, and in our efforts to build war and transport machines it has become necessary to take conservative measures with some elements in particular and with all materials in general. Such changes would probably never be made in other times, or if so, without notoriety. Today, almost everyone has heard of "substitute" or "National Emergency" steels, of synthetic rubber, and of new uses for plastics. The application of new materials to aircraft engines, born of necessity rather than by performance promise, has not been particularly spectacular but is nevertheless of interest to those familiar with the industry's problems. All changes are motivated by the necessity for conserving strategic materials; all must be made with no sacrifice in existing quality and with an open mind to the designer's trend toward increased strength and power.

■ Steel

Aircraft engines, in common with other wartime products using alloy steels, will soon have an almost completely new set of alloy-steel compositions for the majority of working parts. These steels have been selected by the SAE Aircraft Materials and Processes Coordinating Subdivision from the well-known National Emergency series, which has been designed to make our strategic alloying elements – nickel, chromium, molybdenum, vanadium, and tungsten – stretch to cover our rapidly increasing alloy-steel production without sacrificing necessary physical properties. Those NE compositions selected have been incorporated in Aircraft Material Specifications, and are now designated as AMS alternates.

With the exception of AMS 6415 (SAE 4340) steel, aircraft-engine steels have been of the nickel, nickel-chromium, nickel-molybdenum, or chromium-molybdenum types. It is obvious that with the rapid increase in aircraft-engine production, the scrap returned to the steel mills will involve greater and more important tonnage of the above alloying elements. Segregation of this scrap by the manufacturers and the steel mills has become increasingly difficult, with the result that residual chromium and molybdenum in nickel steels, molybdenum in nickel-chromium steels, chromium in nickel-molybdenum steels, and nickel in chromium-molybdenum steels have already shown a rapid increase. Since these residual alloys make none of the desired improvements in the steels containing them, they are, in effect, being wasted. It is the purpose of the NE steels used in aircraft engines, listed in part with their AMS numbers in Table 1, to eliminate the waste of these residual alloys by utilizing to the fullest extent all three of the involved alloying elements, nickel, chromium, and molybdenum. This will make possible a "dilution" of the old steels, with the result that for equivalent physical properties the new steels will require smaller percentages of each alloying element by combining the effects of all three.

In general, the largest saving will be made in nickel, through the addition (plus residuals) of chromium and molybdenum to the nickel steels, molybdenum to nickel-chromium steels, and chromium to the nickel-molybdenum steels. Chromium will be saved by the use of nickel in chromium-molybdenum steels and molybdenum in the nickel-chromium steels. However, the only net savings, in the true sense of the word, will result from the use of residual elements and, in the case of most aircraft alternates, a smaller percentage of total alloy content, offset by the use of molybdenum as in the case of the higher alloy carburizing types. As seen in Table 2, substantial savings in nickel and chromium will be made through the use of the new steels in aircraft engines, but not without a corresponding increase in molybdenum.

Considerable publicity has been given to the NE steels

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 12, 1943.]

Table 1 – [1]Tentative Alternate Steel Specifications – Aircraft Quality Alternate Steel Composition[2]

AMS No.	Form	C	Mn	Ni	Cr	Mo	AMS No. Superseded	Typical Usage
6260	Bar	0.08/0.13 0.08/0.13 0.09/0.14	0.70/0.90 0.45/0.60 0.45/0.60	2.0/2.5 3.25/3.75 4.75/5.25	0.80/1.10 1.40/1.75	0.30/0.40	 6250 6240	All highly stressed carburized gears
6262[3]	Bar	0.08/0.13	0.70/0.90	2.0/2.5	0.80/1.10	0.30/0.40	6252[4]	Carburized crankshaft
6264	Bar	0.15/0.20 0.14/0.19	0.70/0.90 0.45/0.60	2.0/2.5 3.25/3.75	0.80/1.10 1.40/1.75	0.30/0.40	 6254	Carburized starter and accessory drive shaft: supercharger impeller drive shaft
6270	Bar	0.12/0.17 0.11/0.17	0.70/0.90 0.45/0.60	0.40/0.60 1.65/2.00	0.40/0.60	0.15/0.25 0.20/0.30	 6290	Carburized bearing rings and spacers; valve tappet guides
6274	Bar	0.18/0.23 0.17/0.22	0.70/0.90 0.45/0.60	0.40/0.60 1.65/2.00	0.40/0.60	0.15/0.25 0.20/0.30	 6294	Carburized valve tappets, tappet rollers and pins
6280	Bar	0.27/0.33 0.27/0.33	0.70/0.90 0.40/0.60	0.40/0.60	0.40/0.60 0.80/1.10	0.15/0.25 0.15/0.25	 6370	Welded assemblies
6320	Bar	0.33/0.38 0.33/0.38 0.33/0.38	0.75/1.00 0.60/0.80 0.60/0.80	0.40/0.60 1.65/2.00 1.10/1.40	0.40/0.60 .... 0.55/0.75	0.20/0.30 0.20/0.30	 6310 6330	All important studs, nuts and bolts
6322	Bar	0.38/0.43 0.38/0.43 0.40/0.45 0.35/0.42 0.38/0.43	0.75/1.00 0.60/0.80 0.60/0.90 0.70/0.90 0.75/1.00	0.40/0.60 1.65/2.00 1.00/1.50	0.40/0.60 .... 0.45/0.75 0.80/1.10 0.80/1.10	0.20/0.30 0.20/0.30 .... 0.15/0.25 0.15/0.25	 6312 6332 6380 6382	Crankcase sections, valve spring washer and locks, rocker arms general purposes, medium to light sections
6357	Sheet	0.33/0.38 0.30/0.35	0.75/1.00 0.40/0.60	0.40/0.60	0.40/0.60 0.80/1.10	0.20/0.30 0.15/0.25	 6352	Baffle clips and brackets
6530	Tubing	0.27/0.33 0.27/0.33	0.70/0.90 0.40/0.60	0.40/0.60	0.40/0.60 0.80/1.10	0.15/0.25 0.15/0.25	 6360	Oil Transfer Tubes and welded assemblies where heat treatment is necessary
6535	Tubing	0.33/0.38 0.32/0.39	0.75/1.00 0.40/0.60	0.40/0.60	0.40/0.60 0.80/1.10	0.20/0.30 0.15/0.25	 6365	Push rods

[1] Prepared by SAE Aircraft Materials and Processes Coordinating Subdivision.
[2] Silicon 0.20 to 0.35%; phosphorus and sulfur each 0.040% max.
[3] Same as 6260, except for higher hardenability requirements.
[4] Same as 6250, except for higher hardenability requirements.

and their physical properties, usually including tensile strengths, toughness, and so forth. Such data are valuable for metallurgical background, but it is probable that the factors requiring the most urgent attention are related to production requirements in the steel mills, rolling mills, forge shops, and manufacturing plants. It has been amply proved in the past that the ultimate tensile strength, yield point, toughness, ductility, and fatigue strength of all heat-treated steels, regardless of composition, are very closely proportional to hardness, provided that all sections of the steel part have been uniformly hardened throughout in heat-treating.

Given a required set of physical properties, established for aircraft engines by previous experience, it has remained for the metallurgists to choose for application to engine parts those NE steels which would provide uniform hardening throughout all sections of those parts, with a minimum of internal stress induced by the hardening procedure. Water or brine quenching would often produce the full hardening required with plain carbon steels, but to prevent setting up internal stresses, and thus increase the reliability of the parts, a less drastic quenching or hardening action is required along with lower carbon contents. The alloys, nickel, chromium, and molybdenum, in this case, provide the answer, since in the proper proportions they make possible the required full-section hardening of parts by quenching in a milder-acting, hot oil.

The hardenability of a steel is a measure of the depth to which the steel will harden above a chosen hardness level

Table 2 – Estimated Savings *

Nickel	10.85 lb saved
Chromium	6.8 lb saved
Molybdenum	2.42 lb added

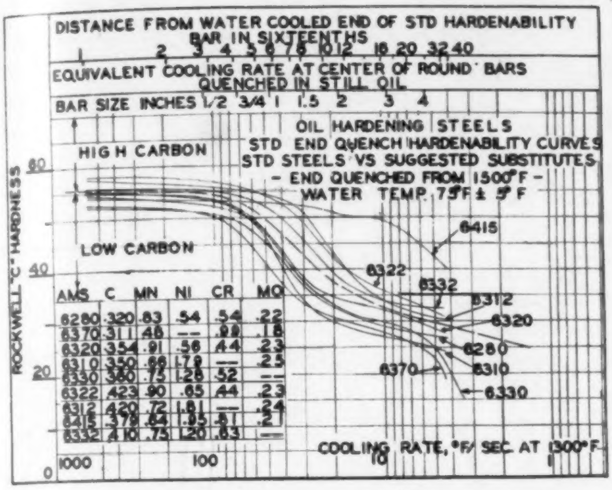
* Estimated savings (neglecting residuals) of nickel and chromium, and the added requirement for molybdenum as a result of the application of alternate AMS steels to aircraft engines. Data are based on one 14-cyl engine.

as a result of a specified quenching or cooling rate. It is apparent that in the selection of new steels for aircraft-engine parts, hardenability (not to be confused with intensity of hardening, which is primarily a function of carbon content) constituted the most important factor in appraising the new steels in terms of the old.

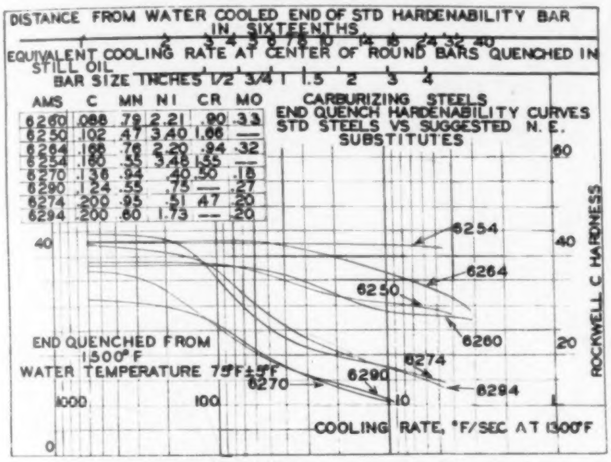
Hardenability of oil-hardening aircraft steels may, of course, be measured by quenching round bars of increasing diameters in oil, and determining from the hardness produced across sections of the rounds, the maximum section which may be fully hardened with each steel considered. However, a more complete measurement of hardenability, and one which is simpler and more reproducible, is made possible by the standard Jominy end-quench method. This method involves water quenching the scale-free end of a 1-in. diameter x 3-in. long bar by suspending the properly heated bar at a fixed distance over a jet of water at 75 F ± 5 F. In this way, a series of known cooling rates are produced, from a maximum at the water-cooled end to a minimum at the opposite end. Hardnesses, determined along the side of the specimen, are correlated with cooling rates, which in turn have been calculated for the centers of various sizes of oil-quenched rounds. Thus, with cooling rate as a parameter, it is possible to compare hardenability of all steels in terms of oil-quenched rounds as well as for other cooling rates.

Typical hardenability curves for most of the AMS alternate steels are to be found in Figs. 1 and 2, along with those AMS steels which the alternates will replace. It will be noted in Fig. 1, which includes the non-carburizing steels, that AMS 6415 steel is outstanding in its hardenability, and since it is a chromium-nickel-molybdenum type it will not require an AMS alternate. AMS 6320 and AMS 6322 show hardenabilities which well represent the large group of steels they will replace, as outlined in Table 1.

The hardenabilities of the four main AMS alternate



■ Fig. 1 – Hardenability curves – oil hardening steels



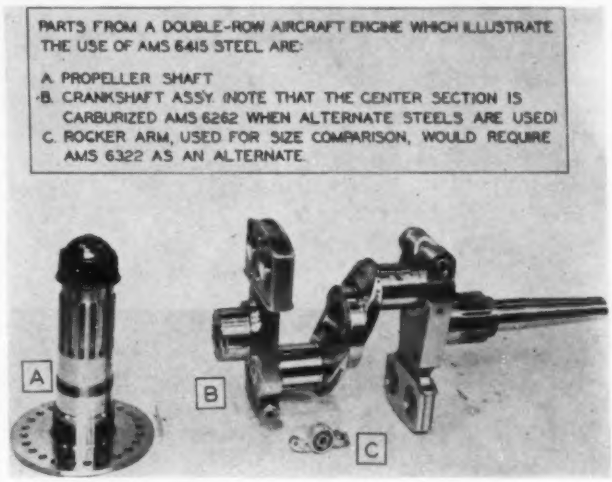
■ Fig. 2 – Hardenability curves – carburizing steels

carburizing steels are very close to those of the old steels, as shown in Fig. 2.

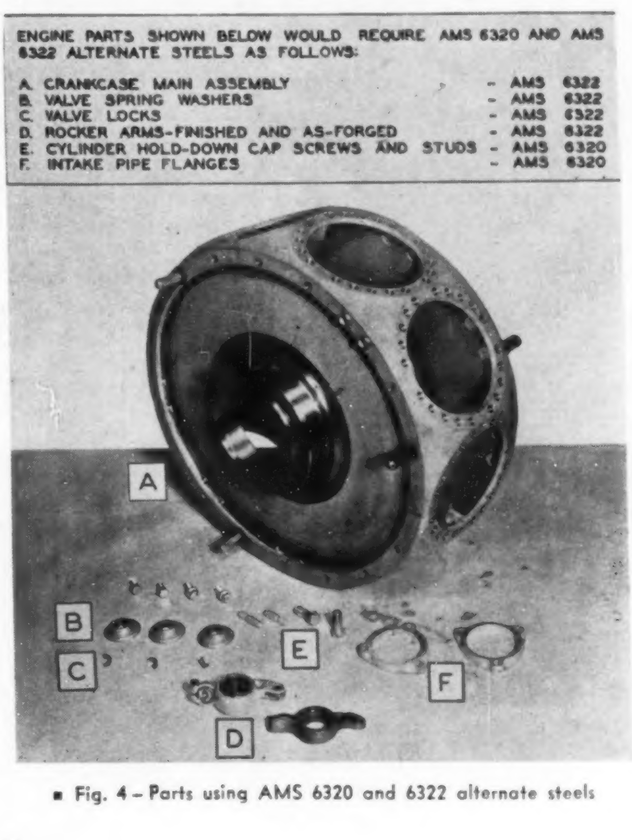
It will be noted that a wide range of hardenabilities has been represented in the hardenability curves illustrated in Figs. 1 and 2 for application to parts of all section sizes. In general, the sharp definition of these groups indicates that a very good job has been done in selecting the AMS alternates.

In order to provide a clearer understanding of the application of depth-hardening power, or hardenability, to actual aircraft-engine parts, groups of these parts have been arranged in Figs. 3 to 6 according to the steels used as outlined in Table 1. In most cases, a comparison of the sizes of the parts and the sections involved may readily be associated with the relative hardenability of the steels used as shown in Figs. 1 and 2. A relationship may also be established between hardenability and total alloy contents from Table 1. Carbon contents should be considered as having a very large effect on hardenability. The crankshaft parts illustrating AMS 6415 steel (excluding the center section) in Fig. 3 are actually heat-treated as forgings for production purposes; hence the very high hardenability required of this application. The crankcase sections illustrating the application of AMS 6322 in Fig. 4 appear large compared to the rocker arm also included, but, as heat-treated in production, the crankcase does not have much heavier sections than the rocker arm that is heat-treated as a forging. The valve spring washers and locks included in this group have light sections but require a comparatively high level of hardness, which may be obtained by the higher carbon content of the AMS 6322 steel. In this connection, it should be remembered that in addition to a mild quench, the freedom from internal stresses demanded of an aircraft part also requires thorough stress relieval or tempering after the quench. Higher carbon and the presence of such alloying elements as chromium and molybdenum allow more efficient stress relieval at higher temperatures without a sacrifice of hardness.

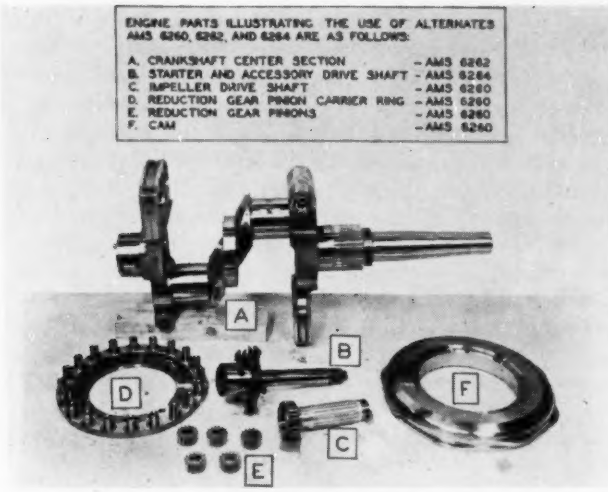
Fatigue strength, which is an extremely important design factor in aircraft-engine parts, has previously been related to hardness and ultimate tensile strength. Figs. 7 and 8 show fatigue curves plotted from data obtained on R. R. Moore rotating-beam machines using representative AMS



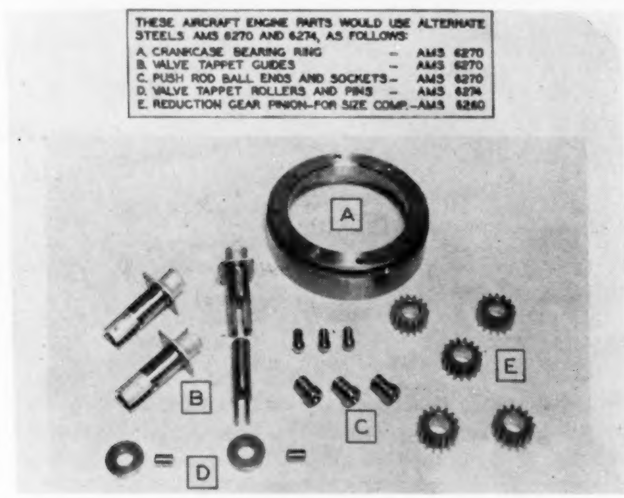
■ Fig. 3 – Parts using AMS 6415 steel



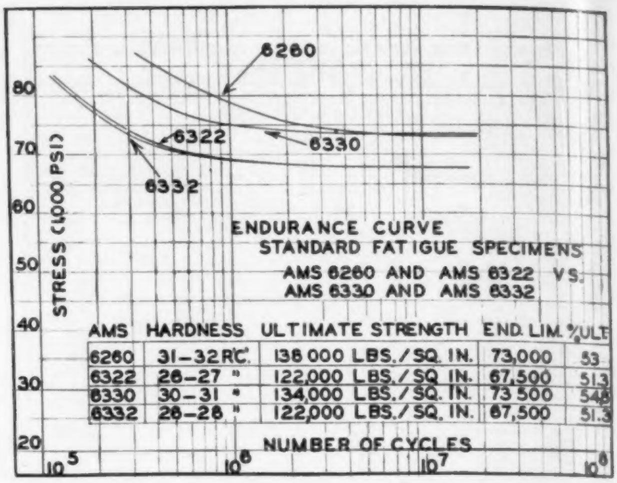
■ Fig. 4 – Parts using AMS 6320 and 6322 alternate steels



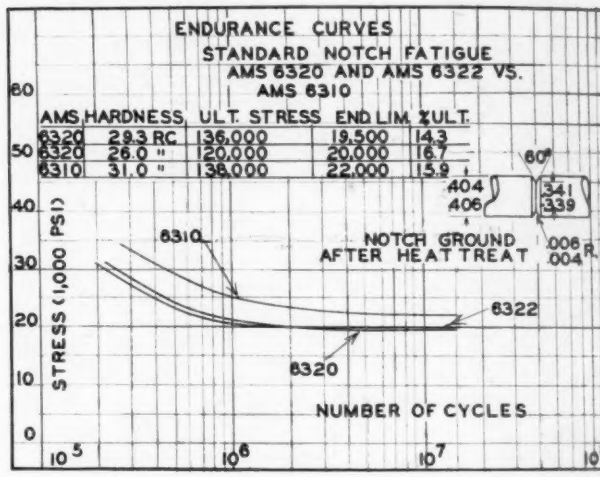
■ Fig. 5 – Parts using AMS 6260, 6262, and 6264 alternate steels



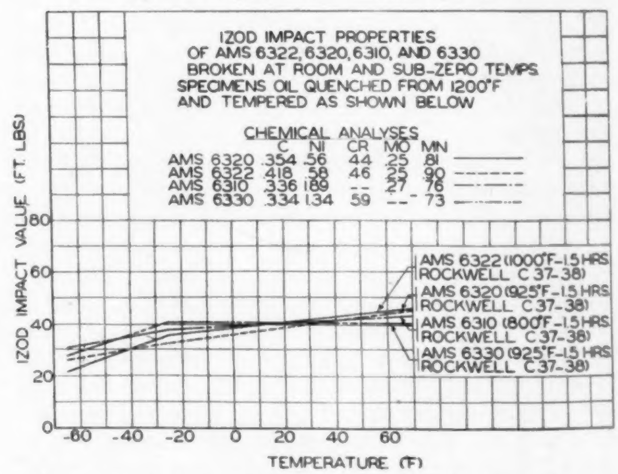
■ Fig. 6 – Parts using AMS 6270 and 6274 alternate steels



■ Fig. 7 – Fatigue curves – standard specimens



■ Fig. 8 – Fatigue curves – notched specimens



■ Fig. 9 – Variation of impact properties with temperature

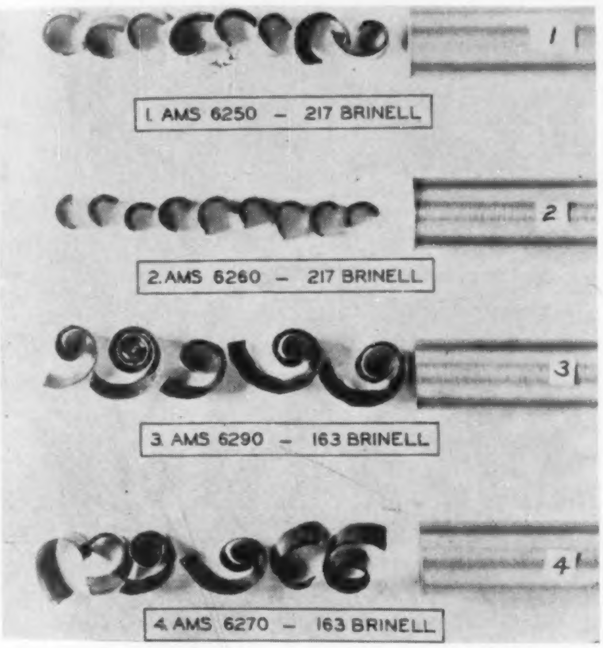
steels and alternates. For the standard polished specimens, the fatigue limits of the new steels have been found consistently to average approximately 50% of the ultimate tensile strength, while the notched specimen (notched after heat treatment) averages about 15% of the ultimate. These figures have been substantiated by many different types of steel tested in the past and compare favorably with the steels they will replace.

Because of the interest in the properties of both present and alternate AMS steels at the subzero temperatures encountered by operation of aircraft at high altitudes, Fig. 9 has been included to show that even as low as –65 F, there is no dangerous drop in the impact properties of either the new or the old steels.

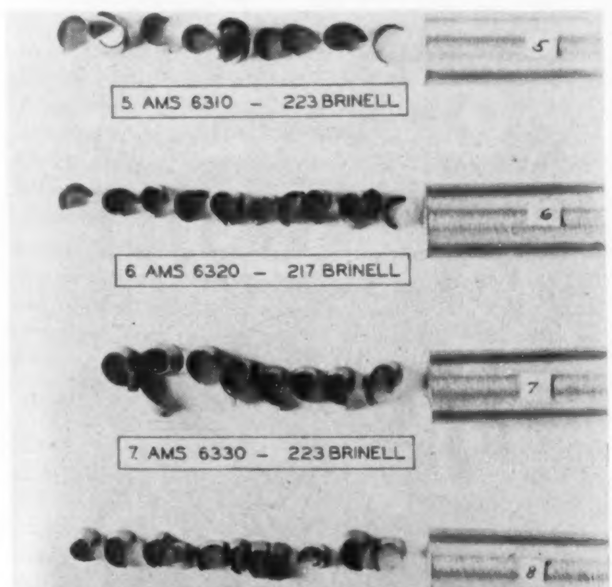
As noted before, it is the adaptability of the alternate steels to production operations that is actually of paramount importance, especially at this time when production stoppages cannot be tolerated. For this reason, one of the most important production factors to be considered in the use of the new steels in aircraft engines is their machineability as compared to the present steels. Tentative machineability tests conducted at the Wright Aeronautical Corp. on bar stock have indicated that the alternate AMS steels in both the carburizing and non-carburizing grades will probably have machining properties equal or slightly superior to the present steels. Tests were run by turning bar stock identically heat-treated to the same hardnesses with standard, superfinished tools at constant feeds and speeds, and comparing finish, tool life, chip formation, and machine tool power input per lb of stock removal. Finishes obtained were approximately equal. Tool life has not been accurately compared as yet, since the tool life on both new and old steels has been too high to measure with the limited supply of alternate steels immediately available.

Power inputs to the machine tool per lb of stock removal for new and old steels appeared to be identical. The alternate steels have shown a slightly more desirable chip formation, as indicated in Figs. 10 and 11 by their smaller and more uniform size. Further tests are required to check the relative machineability of the steels in other machining operations, but it is believed that the lower nickel and higher molybdenum contents of the alternate steels will improve machineability somewhat, and in no case show worse results than with the present steels.

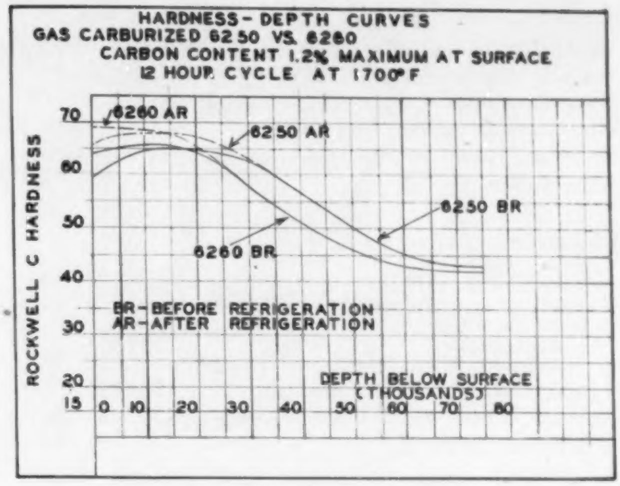
Since most steel engine parts must be heat-treated, such operations may become a serious bottleneck in the production line. One of the most time-consuming and important operations in heat-treating is carburizing for case-hardening



■ Fig. 10 – Chip formations of alternate and present AMS steels



■ Fig. 11 – Chip formations of alternate and present AMS steels



■ Fig. 12 – Hardness-depth curves

applied to all important gears, cams, and shafts, as listed in Table 1. Not only is the time element to be considered but also the case characteristics resulting from the carburizing treatment, which, in addition to being affected by the case carbon content, may vary considerably with various types of steel. In Fig. 12 is shown a case depth-hardness comparison of the cases produced by carburizing and oil quenching AMS 6250 steel and its alternate AMS 6260. From these curves it is seen that the alternate steel, AMS 6260, shows a somewhat slower carburizing rate than the now-used AMS 6250. This difference in carburizing rates is not altogether desirable, but will not cause any important increase in carburizing times in production. To offset the disadvantage of the alternate steel in its lower carburizing rate, it will be seen that a higher surface hardness is made possible by the alternate steel largely because the reduction in nickel and the addition of molybdenum has decreased the austenite retention at the surface. Austenite is a relatively soft constituent in steel, which in this case is retained at room temperature through an incomplete hardening reaction in the steel during oil quenching. Because austenite is soft and ductile, it may easily cause pitting and galling of gears and cams, and is therefore highly undesirable. Of course, in the illustration given in Fig. 12, the carbon content of the case of the two steels has purposely been made to run high in order to compare austenite retention (high carbon contents increase the tendency toward austenite retention), whereas in production the carbon is controlled to a lower figure. The dotted lines indicate the increase in hardness found when the steels were refrigerated to −100 F, under which condition the austenite breaks down, and the hardening reaction is completed. The spread between original and refrigerated hardness curves may be used as a qualitative measure of austenite retention. Since case carbon contents are difficult to control with exactness under production conditions, the decreased tendency of the alternate steels to retain austenite on gear and cam surfaces is highly desirable and important.

Distortion in heat-treatment, which is an ever-present production hazard, especially with aircraft parts involving light and intricate web sections, will probably be about the same with the new steels as with the old, as shown by preliminary tests. This property is usually proportional to the hardenability of the steel regardless of composition.

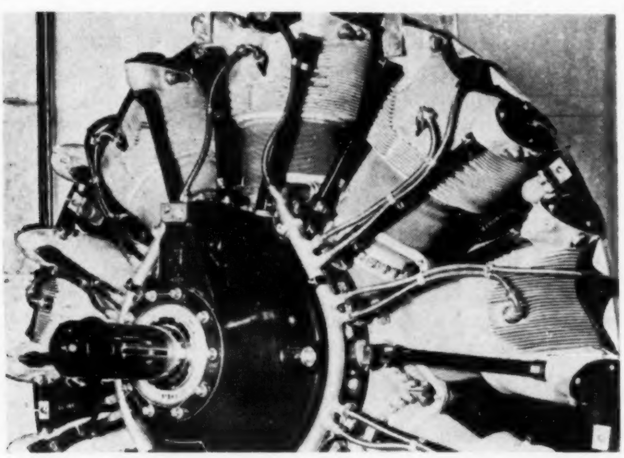
It is believed that in all respects the alternate AMS steels

which have been discussed will be equal to, and in some cases better than, the steels now used in aircraft engines both from the engineering and production standpoints. In addition, as indicated in Table 1, the steel compositions required will be fewer in number, which will allow a welcome "house cleaning" of carry-over and unnecessary compositions.

■ Plastics

Baffles – The baffles on an air-cooled engine deflect the air from its natural course so that it will remove heat from all the fins of the cylinders. These deflectors, when made of aluminum, were cut, formed in several stages, heat-treated, anodized, and covered with three coats of paint. Now made of a cotton fabric impregnated phenolic resin plastic, these baffles give better service and show less breakage than the metal ones because their vibration-damping characteristic improves fatigue life. They are easily formed in one operation of a press and have no other machining difficulties. No paint coat is necessary since a dye is mixed in with the resin compound. Plastic baffles were tested first on single cylinders, then full-scale experimental models before being finally released to service. The saving of aluminum amounts to 29.5 lb for the average engine. The baffles are shown in Fig. 13.

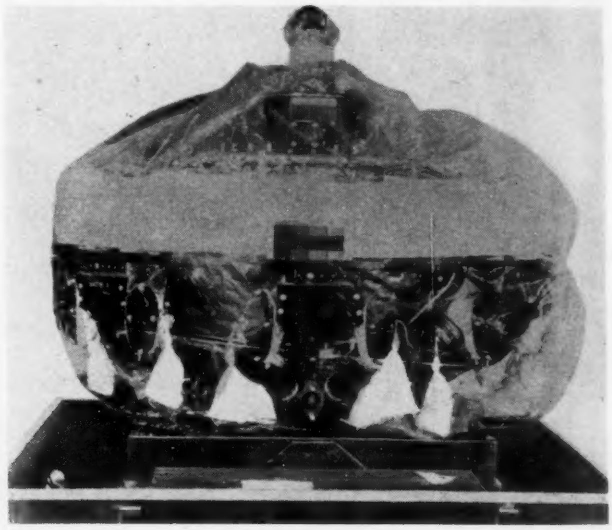
■ Fig. 13 – Comparing the new plastic baffles (left) with the old aluminum baffles (right)



■ Fig. 14 – View of the front of the engine showing plastic push-rod housings

Push-Rod Housings – Push-rod housings are tubes about 12 in. long and 1 in. in diameter which cover and protect the push-rods extending from the cams in the crankcase to the valve assemblies in the cylinder head, as shown in Fig. 14 (front of engine). The housings are also return channels for oil carried to the rocker mechanism. A saving of 4 lb of aluminum is effected, as there are two rods per cyl.

Pliofilm – The use of moisture-impervious, transparent plastic film for dehydration and preservation of engines during shipment and storage is now so extensive that mention of its importance may be made here, although it is not an engine material. Fig. 15 describes more quickly than words the manner in which an engine is hermetically sealed and placed in the packing box. Bags of silica gel dehydrating agent attached to the cylinders maintain a relative humidity in a safe, non-corrosive range. An indicator card, which changes color when the relative humidity has risen above a safe range, is attached within the plastic envelope permitting rapid inspection through a port in the packing box. This is a new use for plastic sheet packaging and, while it does not exactly substitute a previously used material of any importance, it has opened a field of improved practice in storing engines under all conditions.



■ Fig. 15 – View showing how an engine is hermetically sealed before shipment

Plastic plugs containing drying agent also are inserted in engine openings to prevent corrosion of the interior. Such plugs are of the color-indicating type to show how long they are active. Satisfactory reports have come from distant parts, such as Australia and Libya, on the condition of engines shipped by this method.

Aside from the parts just mentioned, the use of plastics in engine construction appears limited because of temperature and stress conditions encountered in a powerplant where the highest output is demanded from the least weight. Many small items will no doubt be developed, such as plastic tubes, to replace neoprene as protection against chafing where ignition shielding leads are attached to the engine, and other small gadgets, but no major material changes to plastics seem likely.

Synthetic Rubber

Natural rubber parts were formerly given light consideration by designers because the control of physical properties was not particularly good. With the introduction of synthetic products, the rubber industry has developed improved properties by a variety of blends to fit individual requirements of heat, oil, and fuel resistance. Failure to perform satisfactorily under these conditions seldom results in failure of the powerplant, but can nevertheless have serious consequences, as in the case of severe oil leakage on a single engine airplane, which could cover the cockpit windshield and obscure the pilot's view.

The invasion of Java and the Malay Peninsula cut off our natural rubber source, and the industry then turned to neoprene synthetic since this material had been available for a number of years. Considerable experience in compounding and processing neoprene made the transition to synthetic products comparatively easy until the demand caused this type to become scarce. New types of synthetic rubber were then promoted and placed on the market. These were the butadiene materials, such as Hycar, Perbunan, Chemigum, and Ameripol. Some of these newer compounds showed characteristics that made them more desirable for certain applications. Where conditions of low swelling in hot engine oil or high heat resistance and low compression set were desirable, corresponding synthetic compounds could be made. This opened the possibilities of compounding and blending to order with all the attendant difficulties of specifications, control of processing, and approvals. Laboratory tests showed the butadiene materials to have a tendency to lose tensile strength and elongation at elevated temperatures, necessitating caution in their use where strength and elasticity were factors in performance.

Research by the rubber vendors has reduced a great many of the processing difficulties, and careful selection of plasticizers, fillers, and curing agents has produced a variety of compounds for specialized applications. Various types, however, will not show relatively similar volume changes or hardening characteristics on contact with straight-run aviation gasoline, aromatic fuels, petroleum solvents, ethylene glycol, de-icer fluids, and special hydraulic system liquids. A good aromatic-fuel-resistant material can be produced from Thiokol (a polysulfide basic polymer), but such a compound would not stand up under contact with lubricating oil or dry heat. It could not be used, therefore, in an oil system with fuel dilution for cold weather starting. One such part requiring the use of this type of material is a molded drain hose connection, as shown in Fig. 16. This part is not only one of the hardest

Fig. 16 – Molded drain hose connection

applications to meet with regard to physical properties, but also a difficult one to mold to dimensions.

An example of heat failure is shown in Fig. 17, where a magneto drive shaft oil seal diaphragm has cracked around the bead allowing oil to escape at the engine rear.

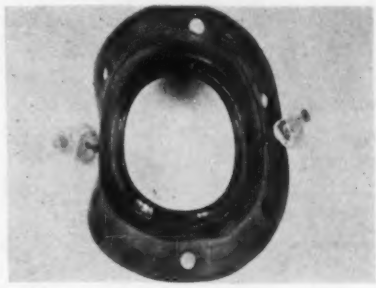


Fig. 17 – An example of heat failure in a magneto drive shaft oil seal diaphragm (small radial cracks on flattened bead)

Fig. 18 shows the location of the diaphragm in the assembly. The design is rather complicated and the part operates under pressure of the spring at high temperature. After some experiments, a synthetic compound was developed to meet the requirements.

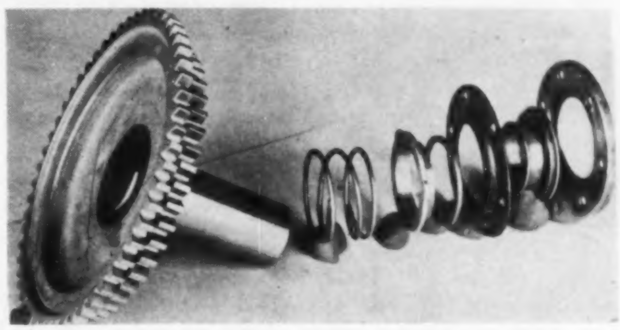


Fig. 18 – Magneto drive shaft oil seal diaphragm assembly

In the theater of war, we find our aircraft in all parts of the globe and stratosphere. Temperature-sensitive materials, such as the rubber type, must now be replaced with substitutes of a broader range. Generally speaking, synthetic rubber parts are hard and brittle at temperatures of —60 F, and vibration can cause them to shatter. A special cold-resisting neoprene has been developed that will remain flexible below —60 F, but its application is limited since it will not withstand aging at temperatures around 150 F in either oil or air, as all the freeze-resisting plasticizing oils will be leached out during the oil immersion. Considerable research is being done at the present time to find materials for air deflector attachments that will withstand the extremes in operating conditions. The development of fire-resistant fuel hoses within the last year has added to the safety of aircraft operation, not to mention self-sealing fuel tanks for military craft. The progress made by the synthetic rubber industry has been quite rapid and in step with the higher performance requirements.

Miscellaneous

Silver is now in satisfactory use as a bearing material. Master rod bearings, formerly of copper-lead, now of silver, exhibit higher fatigue strength, better heat conductivity and greater capacity to withstand higher loads. This means higher dive speeds can be attained, for the tendency of the

bearing to deform under these conditions is reduced. Bearings made of the new material are no more expensive than the old in spite of the precious-metal aspect. After the problem of poor bond was overcome, it was found that uniform silver bearings could be produced quite easily on a number of base materials. X-ray and ultra-violet ray inspection are used in quality control. A problem of production has also been eased by the use of this material instead of copper-lead, which is extremely difficult to produce to aircraft-engine standards.

Higher performance at high altitudes has shown that the ignition system is suffering under greater electrochemical stresses than ever before, and more care in the selection of materials indicates a trend toward inorganic compounds. There has been substantial development in the field of ignition harness fillers for air exclusion; ceramics for spark-plug insulation as well as for small sleeves where the ignition cable is attached to the spark plug; and a glass-mica compound for the elbow attaching fixture.

Metal surfacing is finding an increasing number of applications for a variety of purposes. Coatings to prevent chafing or fretting between adjacent steel surfaces have eliminated failures in certain parts. These are applied by electroplating, as in the case of silver, or by solution attack as in the case of phosphate treatment. The latter provides a dull black, lightly etched finish that is quite effective against the mysterious powers that govern chafing. A sulfur treatment which darkens the surface of sintered bronze clutch plates from the formation of copper sulfide has shown an improvement in the break-in quality of these parts and prevented welding of the plates under severe operation. The coating is easily applied by exposure to sulfur-bearing compounds in gaseous or liquid form. Hot aluminum spray is now used extensively on aircooled cylinders to replace paint, with the result that corrosion resistance has been considerably improved for engines operating near salt water. In addition, it does not have to be replaced as often as paint at overhaul. Zinc shows promise as a substitute for cadmium in plating a great many engine parts as protection against corrosion either while the part is being made in the shop or as externally located on the engine. Sufficient service reports have not yet come in to state definitely that this less expensive and less strategic metal will successfully replace cadmium. Zinc-coated lockwire as a replacement for stainless steel has been tried but has given trouble when the soft zinc is skinned from the wire at installation by pulling through studs. If so used internally, the small zinc chips could remain in the engine and cause damage.

Several new alloys in the non-ferrous field have been given a trial. A substitute aluminum alloy for cast cylinder-head material (AMS 4220) was cast experimentally. In this material the 2% nickel content was replaced by iron. By laboratory test, the properties of the new material were satisfactory, and the foundry reported casting properties equivalent to the old; but the engine said "No!". Every cylinder head made of this material cracked on experimental running – proof again that substitution on major parts cannot always be made on paper or in the laboratory.

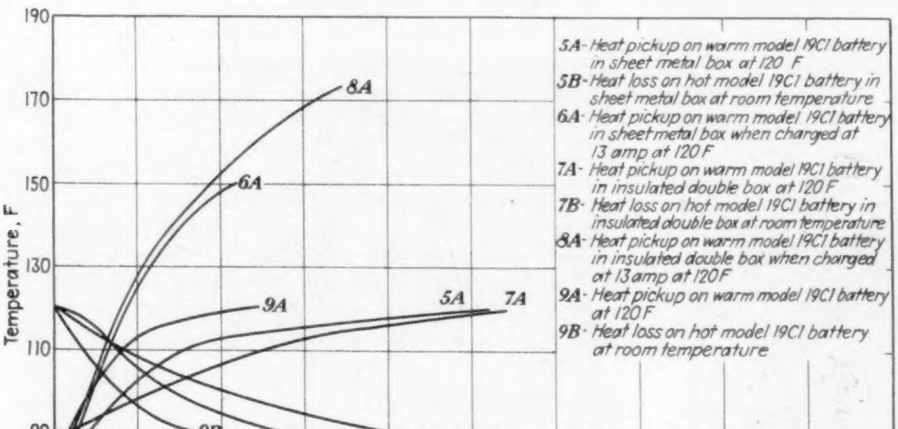
The changes and substitutions described herein show that the field for new materials is now limited; demands are dictated by production requirements rather than by performance; and supply must meet demand under emergency conditions. The post-war era will no doubt find many new ideas and designs ready for experiment, and the metallurgist will again be challenged to find the material.

Storage Battery Performance at Low Temperatures

continued from page 156

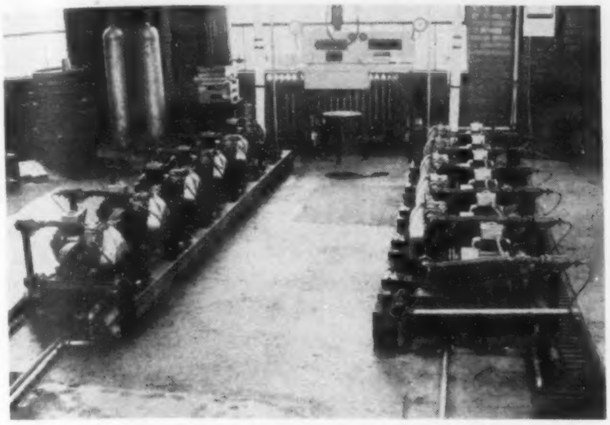
While it has no particular bearing on the cold behavior of batteries, Fig. 34 has been added to show the warm-up on a 4H battery from approximately 80 F in an ambient of 120 F, together with the behavior when charging under these conditions. These data are more useful for hot-climate operation. The use of the battery in an insulated box exhibits some advantages in this case, as the warm-up is retarded over that of a bare battery during a running day, and at night when it cools off the battery box could be opened up.

In conclusion, the battery installation must be adapted to the performance required under a given set of temperature and engine conditions. It should be insulated or kept warm to be usable, and if allowed to get too cold, it must be warmed up. It must be kept in a good state of charge to prevent freezing. It must be further warmed while the vehicle is in operation, to permit it to be charged.



■ Fig. 34 – Heat-loss and heat-pickup curves – Army-type battery 4H – room temperature to 120 F

THE INFLUENCE of LUBRICATING OIL VISCOSITY on CYLINDER WEAR



■ Fig. 1 – The batteries of single-cylinder engines used in the wear tests

by H. A. EVERETT

Head of the Department of Mechanical Engineering
The Pennsylvania State College

RESULTS of tests to determine the influence on ring and cylinder wear of one single property of lubricating oil, namely viscosity, are reported by the author.

Prof. Everett measures the amount of iron wear in the cylinders by evaluating the iron contamination of the crankcase oil used, adjusted to allow for the oil that has been lost by consumption, leakage, and samples.

The results of Prof. Everett's tests show clearly that cylinder and ring wear, and incidentally oil consumption, in well-lubricated engines decreases progressively with increasing viscosity throughout the range tested, the wear being almost inversely proportional to the kinematic viscosity.

★ ★ ★

THE AUTHOR: H. A. EVERETT (M '36) joined Pennsylvania State College in 1922 as professor of thermodynamics, following a diversified engineering career. Since 1931 he has been head of the Department of Mechanical Engineering. After graduation from Massachusetts Institute of Technology in 1902 with the degree of B. S., Prof. Everett worked as an outside machinist for the Fore River Ship & Engine Building Co.; several years later he went with the N. Y. Shipbuilding Co., as estimator and computer. He then became associate professor of naval architecture at Massachusetts Institute of Technology and, in 1915, was appointed professor of marine engineering in the post-graduate department of the U. S. Naval Academy at Annapolis, Md. Before going to Penn State, Prof. Everett served as chief engineer for the Union Shipbuilding Co., Baltimore, Md.

WEAR of piston rings and cylinder walls is the greatest single item of wear in the modern automotive internal-combustion engine. While under normal service conditions such mechanical wear is slight, the process is continuous and is materially influenced by such factors as (a) operative conditions, (b) characteristics of fuels and lubricants, and (c) the composition and surface condition of the parts themselves. It has been shown[1] that different lubricating oils have a marked influence on cylinder and ring wear, even though scuffing or seizure does not develop; and these tests were undertaken with the purpose of determining the influence on wear of one single property of the lubricating oil, its viscosity.

Special precautions were taken to eliminate all other variables, and the range of viscosities covered was as extensive as was compatible with this restriction. Tests were made with six oils ranging from a viscosity lighter than SAE 10 to a viscosity greater than SAE 70.

Each oil was tested in a battery of six single-cylinder engines operating under conditions artificially controlled to simulate closely moderately heavy-duty road operation of automobile passenger-car engines. (See Fig. 1.)

It was essential in selecting the oils to exclude differences in crude source or methods of refining, which in themselves might influence performance; hence all the oils run were blends of conventionally refined bright stock and neutral from one crude source (Pennsylvania). By blending these in different proportions, oils of various viscosities could be obtained, each blend retaining the family characteristics peculiar to the lubricants of one particular field. Both bright stock and neutral were conventionally refined Pennsylvania oils, and the blending was done in the Petroleum Refining Laboratory of The Pennsylvania State College. A 180 neutral* was used as the light blending agent throughout, except for the lowest viscosity oil.

Data concerning the physical characteristics of each of the blends are given in Table 1.

The engines were of the L-head type, aircooled, 1¾-in. bore by 1⅞-in. stroke**. The pistons, cylinder heads, and connecting rods were of aluminum. Each piston was fitted

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

[1] See The Pennsylvania State College Engineering Experiment Station Bulletin No. 44, 52 pp., 1939 (first published, 1935): "Performance Tests of Lubricating Oils in Automobile Engines," by H. A. Everett and F. C. Stewart.

* 180 neutral denotes an oil having 180 viscosity (S.U.S.) at 100 F; similarly for 75 neutral.

** Made by the Lauson Co.

Table 1 – The Test Oils
Physical Data on Unused Oils

Oil Number	V-6	V-4	V-1	V-2	V-3	V-5
SAE No.	Less than 10	10	20	30	50	70
Gravity, deg API	34.0	30.3	29.7	28.6	27.4	26.0
Viscosity:						
S.U.S. at 210 F	37.06	45.56	49.8	63.6	91.55	152.6
Kinematic at 210 F, centistokes	3.25	5.92	7.23	11.24	18.42	32.40
Absolute at 210 F, centipoises	2.599	4.856	5.905	9.329	15.288	27.39
S.U.S. at 100 F	74.84	183.0	246.8	476.0	1011.8	2395
Kinematic at 100 F, centistokes	14.38	39.4	53.35	103.0	219.0	516.3
Absolute at 100 F, centipoises	12.07	33.86	46.02	89.47	191.73	457.83
Flash Point, F	325	420	430	450	470	535
Fire Point, F	375	475	485	505	530	615
Pour Point, F	+60	+25	+25	+30	+25	+25
Color, ASTM	2	2½	4½	7	8+	8+
ASTM V.I.	104	102	103	103	100	100
Gravity Index	104	98	99	102	104	106

Note: Oils V-1, V-2, V-3, V-4, and V-5 were blends of 180 neutral and bright stock. oil V-6 was 100% of 75 neutral.

with two tapered cast-iron compression rings and one cast-iron oil ring. Lubrication was by the splash system, with an oil sump capacity of approximately ¾ pt, no filter.

A small steam-heating coil was in each crankcase to maintain the desired oil temperature. Cylinder cooling was by means of a fan cast integral with the flywheel located at the front of the engine. A shutter was constructed to fit over the air entrance of the fan housing, so that the quantity of air passing over the cylinder could be regulated. Baffles were constructed around each cylinder for more uniform cooling. A more detailed description may be found in other articles[2, 3].

Each engine was direct-connected to a 1/3-hp synchronous motor that maintained constant engine speed. When driven by the engine, the motor acted as a generator.

The crankcase oil temperature was determined by a thermocouple located in the crankcase and completely submerged in oil at all times. Another thermocouple was located at the rearmost cylinder-head bolt to give cylinder-head temperatures. Previous tests showed that the temperature observed on this rear bolt was approximately equal to an average head temperature taken under the four bolts.

Particular care was taken to maintain these important temperatures constant throughout, approximately 230 F for the crankcase temperature and 450 F for the cylinder-head temperature.

■ Tests

Two groups of six engines each were used. Two oils were thus tested at the same time, one group testing one oil and the other group testing a second oil. Runs were of 8-hr duration.

Before the start of each test, the engines were completely overhauled and flushed out with kerosene. New rings were installed and the engines motored for a brief interval with the heads removed to make sure the rings were functioning properly. The engines were then drained overnight. Next morning, the six engines of each group were filled with a test oil and motored for approximately 30 min with crankcase oil temperature at 230 F. The crankcases were then drained and each group of six was charged with its test oil.

During each test, the engines were operated under full throttle at a constant speed of 1880 rpm, at approximately 0.6 hp, and 56 psi bmep. The cylinder-head temperatures were held at 450 F, and the crankcase temperatures were maintained at 230 F.

Each test consisted of an 8-hr run. Readings of the head temperatures, crankcase temperatures, and rpm were taken every hour. At the end of 4-hr, the engines were momentarily stopped and sufficient oil added to bring the oil to the original level in the crankcase. (In the test on the lightest oil, the additions were made every 2 hr.) At the end of the test, the crankcase was drained. This residual used oil was analyzed as it came from the crankcase without the addition of any make-up oil.

Non-leaded gasoline was used throughout all tests.

■ The Determination of Wear

The greatest iron wear in internal-combustion engines occurs in the cylinders. Iron particles worn from an engine during normal service are extremely small. Actual photomicrographs show that these particles, for well-lubricated wear, vary in size from 1 to 5 microns. These small particles remain suspended in the oil almost in a colloidal state, hence all the iron worn from the walls and rings is in the residual lubricating oil, except that which leaves the engine in the oil lost by leakage, samples, and exhaust gases. Thus, a quantitative evaluation of the iron contamination of the crankcase oil can be used to estimate total iron wear.

This method of evaluating cylinder and piston-ring wear by measuring the iron contamination of the crankcase oil has been described previously[1, 4]. It has been in successful use by us for the past seven years, and a similar method was employed by Boerlage[5] for determining iron wear in large marine diesel engines. The method is extremely sensitive and lends itself admirably to an investigation such as this, as it obviates the necessity of the very long periods of operation required for comparisons by mechanical measurements.

Briefly reviewed, the method consists of taking a 10-g sample of the used crankcase oil and burning this to a residual ash. This is then dissolved in hydrochloric acid, diluted, and ammonium sulfocyanide added. The resulting solution is reddish in color. The intensity of the color depends on the amount of iron. The sample solution is then matched colorimetrically with other solutions of predetermined iron content. A quantitative evaluation results. The filter photometer method employed by the Petroleum Refining Laboratory admits a sensitivity of 1 part in 50,000,000 and an accuracy of $\pm 0.2\%$[6].

The iron content of the residual oil as determined above is then adjusted to allow for the oil which has left the

[2] "The Testing of Lubricating Oil Stability in Small Engines," by H. A. Everett and G. H. Keller, presented at the SAE World Automotive Engineering Congress, New York, May 23, 1939.

[3] See The Pennsylvania State College Engineering Experiment Station Bulletin No. 48, 35 pp., 1939: "A Laboratory Method for Evaluating the Influence of Lubricating Oils on Carbon Deposition in Internal-Combustion Engines," by H. A. Everett and G. H. Keller.

[4] See Institution of Mechanical Engineers, Proceedings of the General Discussion on Lubrication and Lubricants, Vol. 1, pp. 451-456: "Rating Oils for Cylinder Wear by the Iron Contamination Method," by H. A. Everett and G. H. Keller (American reprint by ASME, May, 1938.)

[5] See *The British Motor Ship*, Vol. 13, No. 150, Aug., 1932, pp. 171-173: "Cylinder Wear in Diesel Engines," by G. D. Boerlage and B. J. J. Gravestyn.

[6] See *Industrial and Engineering Chemistry*, Analytical Edition, Vol. 8, No. 4, July 15, 1936, pp. 242-244: "Photometric Determination of Iron in Used Engine Oils," by A. R. Rescorla, E. M. Fry, and F. L. Carnahan.

Table 2 – Detailed Operating Data

Test Oil No.	Engine No.	Temperatures, F: Head	Temperatures, F: Crankcase	Weights of Oil, lb: Charged	Weights of Oil, lb: Added	Weights of Oil, lb: Residual	Weights of Oil, lb: Consumed	Fuel Consumption, cc per min	Speed, rpm
V-6 75 Neutral	7	448	227	0.523	0.294	0.408	0.409	6.04	1870
	8	446	230	0.506	0.309	0.423	0.392	5.09	1870
	9	447	231	0.493	0.194	0.465	0.222	4.98	1882
	10	445	229	0.545	0.174	0.426	0.293	5.19	1877
	11	434	234	0.530	0.188	0.448	0.270	5.06	1874
	12	446	234	0.500	0.166	0.396	0.270	4.89	1883
V-4 180 Neutral	7	454	231	0.587	0.000	0.451	0.136	6.87	1885
	8	462	231	0.505	0.037	0.414	0.128	5.53	1893
	9	452	229	0.487	0.084	0.482	0.089	6.00	1900
	10	448	231	0.497	0.067	0.489	0.085	5.19	1884
	11	449	233	0.574	0.000	0.353	0.221	5.15	1885
	12	453	234	0.566	0.077	0.419	0.224	4.86	1890
V-1 SAE 20	1	448	233	0.502	0.062	0.438	0.126	5.11	1885
	2	451	230	0.490	0.102	0.373	0.219	5.12	1885
	3	443	235	0.471	0.105	0.440	0.136	6.71	1870
	4	449	235	0.475	0.119	0.403	0.191	4.88	1876
	5	450	230	0.405	0.190	0.445	0.150	6.23	1878
	6	437	229	0.511	0.121	0.427	0.205	....	1880
V-2 SAE 30	7	451	229	0.443	0.055	0.376	0.122	5.35	1852
	8	451	224	0.422	0.097	0.372	0.147	6.50	1880
	9	445	229	0.505	0.026	0.432	0.099	6.12	1895
	10	451	234	0.473	0.089	0.438	0.124	5.80	1875
	11	439	229	0.493	0.059	0.413	0.139	5.40	1880
	12	437	234	0.480	0.082	0.426	0.136	5.29	1880
V-3 SAE	1	447	233	0.492	0.091	0.472	0.111	5.98	1890
	2	448	232	0.421	0.126	0.489	0.058	5.18	1880
	3	456	224	0.443	0.000	0.406	0.037	6.20	1885
	4	448	233	0.463	0.000	0.414	0.049	5.13	1860
	5	447	228	0.442	0.158	0.362	0.238*	5.91	1880
	6	452	231	0.600	0.000	0.523	0.077	5.81	1870
V-5 100% Bright Stock SAE 70	1	453	234	0.538	0.000	0.493	0.045	5.49	1887
	2	446	238	0.484	0.243	0.370	0.357*	5.29	1880
	3	454	226	0.535	0.000	0.515	0.020	5.75	1878
	4	452	230	0.415	0.082	0.468	0.029	5.06	1882
	5	450	226	0.506	0.042	0.385	0.165*	5.51	1876
	6	452	229	0.472	0.000	0.462	0.010	7.20	1872

* Leakage excessive.

crankcase by consumption, leakage, and samples, and a value determined for the total iron worn from the cylinder and rings, which is termed the *total iron wear*. This adjustment from the iron content of the sample to the total iron wear is as follows:

The iron concentration in the successive amounts of oil leaving the engine by consumption, leakage and samples varies from zero at the start to that found in the residual oil at the end of the test, but if the rate of wear is constant (and experimental evidence confirms this, except on starting) there results a very simple expression which closely approximates the total wear in terms of the iron content of the residual sample and the various oil weights which have left the crankcase.

If no samples are taken from the crankcase during the run:

$$\text{Total Wear} = AX_a + LX_l + BX_b \qquad (1)$$

where:

A = Weight of test oil remaining in the crankcase

X_a = Fraction of iron in test oil remaining in the crankcase

L = Weight of oil that leaked out during the run

X_l = Average fraction of iron in oil leakage

B = Weight of oil burnt or consumed during the run

X_b = Average fraction of iron in oil so consumed

Assuming a constant rate of wear, X_l and X_b become equal to $X_a/2$, and expression (1) simplifies to:

$$\text{Total Wear} = AX_a + (L + B)\frac{X_a}{2} \qquad (2)$$

$$= \frac{X_a}{2}(2A + L + B)$$

The above expression takes care of any reasonable leakage. In cases of excessive leakage (and of abnormally high consumption), the freshening effect of the new oil added is appreciable and renders such runs less reliable. Fortunately in these tests the number of such was very small.

■ Results

Detailed operating data from each of the engines and all of the tests are presented in Table 2 and from these the average data for all six engines for each oil are assembled in Table 3.

Table 3 – Averages of Table 2

Test Oil No.	Temperatures, F: Head	Temperatures, F: Crankcase	Oil Consumption, lb	Fuel Consumption, cc per min	Speed, rpm
V-6 75 Neutral	444	231	0.314	5.21	1876
V-4 180 Neutral SAE 10	451	231	0.149	5.52	1888
V-1 SAE 20	446	232	0.171	5.61	1879
V-2 SAE 30	446	230	0.128	5.74	1877
V-3 SAE 50	450	230	0.066	5.70	1881
V-5 SAE 70 100% Bright Stock	451	231	0.026	5.72	1880

Similarly, for each engine and all tests the *iron content* of the residual oil and the *total wear* determined therefrom are listed in Table 4.

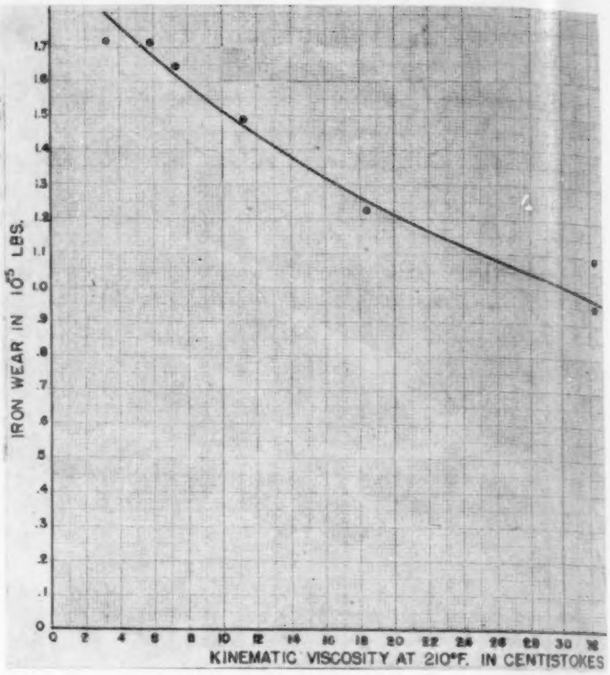
Table 4 – Values of Iron Content and Total Wear

Test Oil No.	Engine No.	Iron Content of Residual Oil, % (multiply by 10^{-3})	Total Iron Wear, lb (multiply by 10^{-5})
	7	2.67	1.631
V-6	8	2.54	1.571
75	9	2.09	1.205
Neutral	10	2.59	1.484
	11	4.88	2.845
	12	2.91	1.544
	7	2.54	1.318
V-4	8	2.89	1.380
180	9	2.47	1.300
Neutral	10	3.05	1.575
SAE 10	11	5.85	2.711
	12	3.76	1.996
	1	2.40	1.203
	2	2.07	0.979
V-1	3	3.68	1.870
SAE 20	4	3.23	1.608
	5	4.00	2.080
	6	3.72	2.150
	7	3.38	1.476
	8	4.66	2.074
V-2	9	3.09	1.488
SAE 30	10	3.41	1.704
	11	1.98	0.955
	12	2.51	1.241
	1	2.37	1.307
	2	2.31	1.197
V-3	3	3.34	1.420
SAE 50	4	2.37	1.038
	5	2.45	1.182
	6	....	.....
	1	1.84	0.948
V-5	2	1.21	0.670
100%	3	1.95	1.025
Bright Stock	4	1.80	0.868
SAE 70	5	1.41	0.658
	6	3.27	1.526

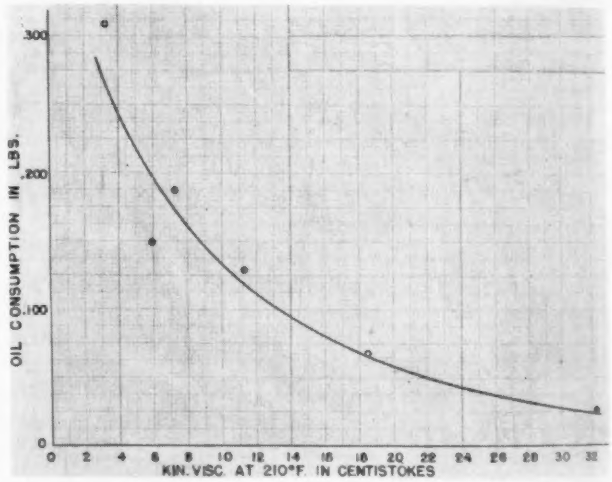
Table 5 – Viscosities of Oils Used and Average Values of Iron Wear

Test Oil No.	Viscosities: Kinematic, Centistokes 100 F	210 F	Absolute, Centipoises 100 F	210 F	Saybolt, S.U.S. 100 F	210 F	Total Iron Wear$\times 10^5$, lb
V-6 75 Neutral	14.38	3.25	12.07	2.599	74.84	37.06	1.713
V-4 180 Neutral SAE 10	39.40	5.92	33.86	4.856	183.0	45.58	1.714
V-1 SAE 20	53.35	7.23	46.02	5.905	246.8	49.8	1.648
V-2 SAE 30	103.0	11.24	89.47	9.329	476.0	63.6	1.489
V-3 SAE 50	219.0	18.42	191.73	15.288	1011.8	91.55	1.229
V-5 Bright Stock SAE 70	518.3	32.40	457.83	27.39	2395.	152.6	0.949 (1.092)*

* If data of units No. 2 and No. 5 be excluded.



■ Fig. 2 – Iron wear versus kinematic viscosity of lubricating oil



■ Fig. 3 – Oil consumption versus kinematic viscosity of lubricating oil

In Table 5 the average values of iron wear are given for each test together with the initial viscosities of the oils used. These latter are expressed in three sets of units and for two temperatures.

A graphical presentation of the data of Table 5 using the viscosities at 210 F is given in Fig. 2. In obtaining the averages for wear, the data from all engines were used. This curve clearly indicates the influence viscosity has on cylinder and ring wear under what might be termed normal operative conditions and continuous operation.

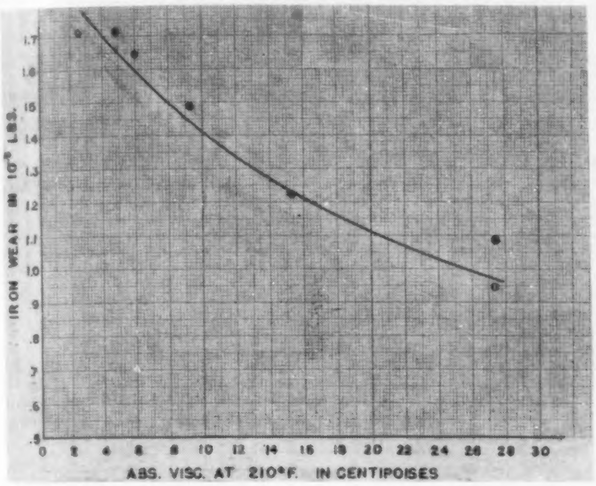
The amount of oil consumed is also influenced by viscosity as may be seen by Fig. 3, drawn from the data of Table 3. For this curve, the data from engines 2 and 5 with the heavy oil, V-5, were disregarded in determining the averages.

The character of the wear curve, that is, decreasing with increasing viscosity, is quite definite but probably underestimates rather than overestimates the simple viscosity influence because of the collateral effect viscosity has on oil consumption. The larger the oil consumption, the greater the freshening effect of the make-up oil added and consequently the smaller the iron contamination in the residual oil. Hence the larger consumptions associated with the lower viscosities have, in turn, brought about by simple dilution a lower iron content in the residual oil than would have been the case if the consumption had been smaller. While the method for computing wear as outlined earlier largely compensated for the influence of this dilution, it is probable that the correction is underestimated for large consumptions.

It should be borne in mind that all of the wears reported are very slight, in no case was scuffing or noticeable abra-

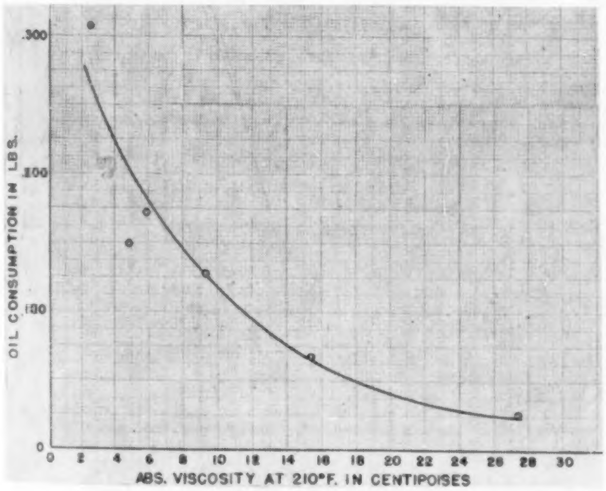
sion evident, and the internal surfaces were all comparable to the cylinder interiors usually encountered in well-maintained engines. Nevertheless, it is evident that the lower the viscosity the higher the wear even though the maximum reported may be well within the limit considered acceptable for good service life.

While kinematic viscosity is probably the preferential basis for comparison, another curve is shown in Fig. 4 in



▪ Fig. 4 – Iron wear versus absolute viscosity of lubricating oil

which the wear data are plotted on the basis of absolute viscosity, and Fig. 5 gives the oil consumption plotted on this basis.



▪ Fig. 5 – Oil consumption versus absolute viscosity of lubricating oil

▪ Conclusions

From these results, it appears that the cylinder and ring wear in well-lubricated engines decreases progressively with increasing viscosity throughout the range tested, the wear being almost inversely proportional to the kinematic viscosity. This is in harmony with our current understanding of fluid film lubrication as the film thickness for a given loading is probably nearly proportional to the kinematic viscosity, and certainly wear is an inverse function of the film thickness. An accurate determination of the character of the lubrication, that is, whether of fluid-film or boundary type, cannot be made, but the small amount of wear, the smooth surfaces, and the light loading due to ring pressures indicate that through the major part of the reciprocating cycle, fluid-film action was predominant.

▪ Acknowledgment

The experimental work was done in the Mechanical Engineering Laboratory of The Pennsylvania State College with the assistance of B. H. Garcia, Jr. and R. E. Ebel, seniors in the Department of Mechanical Engineering. The preparation of the several oils used in the tests, together with the inspection and analytical work on the oils after usage, was done in the Petroleum Refining Laboratory at the College by V. J. Lettieri. To these gentlemen, for their direct operative assistance, and to the administrative staff of the Petroleum Refining Laboratory under Dr. M. R. Fenske for many helpful suggestions, I wish to record my sincere appreciation.

DISCUSSION

Author's Results Applied to Road-Test Data

— H. C. Mougey

Technical Director
Research Laboratories Division, General Motors Corp.

THE data presented by Prof. Everett are very interesting and valuable, and the conclusions he has drawn are sound. However, I do not believe that the SAE viscosity grade recommended for use in a specific engine should be based on Prof. Everett's conclusions.

If the total iron wear, as given in Table 5, is taken as a direct measure of cylinder wear, some additional very interesting conclusions may be drawn. If the amount of cylinder wear with the SAE 20 oil is taken as a standard and assigned a value of 100%, then the amount of cylinder wear with the oils of the other viscosity classification would be as follows:

SAE 30	90%
SAE 20	100%
SAE 10	104%
75 Neutral	104%

These differences in wear are very small and they certainly are beyond the limits of experimental error of any wear test data from ordinary performance in service, and perhaps they are even beyond the limits of experimental error of a road test carried out under accurately controlled conditions. However, it may be of interest to use these percentages in connection with a very elaborate and carefully conducted road test that was made recently. In 1940 a large oil company made a road test in which it operated 12 cars of three different manufacturers about 110,000 miles each, or a total of about 1,250,000 miles[1]. The oil was 20 W. The cars were driven at an average speed of 50 mph, with a maximum speed of 55 mph. The average cylinder wear for all 12 cars was 0.0008 in. per 110,000 miles. If the percentage figures just given are applied to this average cylinder wear value of 0.0008 in., then the average cylinder wear per 110,000 miles to be expected with the other viscosity grades would range from 0.00072 to 0.00083 in. These figures indicate that over the range in viscosity from 75 neutral (75 sec at 100 F) to SAE 30 oil, the difference in cylinder wear per 110,000 miles due to viscosity differences would be approximately one-tenth of a thousandth of an inch. Under ordinary service conditions the rate of cylinder wear is evidently due largely to other factors than the viscosity of the oil. This indicates that the difference in the rate of cylinder wear over the range of SAE 10 to SAE 30 is sufficiently small, due to oil viscosity differences, so that the choice of the SAE viscosity to use in an engine will depend upon other factors, such as oil consumption, the decreased engine friction due to lower-viscosity oils, or the greater ease of starting in cold weather with low-viscosity oils.

[1] See *Automotive Industries*, Vol. 82, No. 10, May 15, 1940, pp. 450-451, 489: "Revealing Tests Produce New Fuel Data," based on a talk by Dr. Thomas G. Delbridge.

TECHNICAL DEVELOPMENTS i

by WILLIAM SCHROEDER
and THOMAS H. HAZLETT

Research Engineers, Lockheed Aircraft Corp.

THE paper deals particularly with those phases of aircraft production concerned with sheet metal forming, namely, basic analyses of sheet metal forming operations, classification of parts into basically similar groups, forming techniques, and limits to which the commonest materials may be successfully formed. Emphasis is placed upon the need for quantitative knowledge of the forming limits for die and aircraft design.

All of the common types of forming equipment, with their applications and limitations, are discussed. Methods capable of high rates of production, such as the rubber pressure hydro-press and the double-acting press, are discussed in more detail. Forming limits are presented and techniques are discussed for flanging, stretching, drawing, and redrawing.

• • •

THE AUTHORS: WILLIAM SCHROEDER, senior research engineer, Lockheed Aircraft Corp., holds Master's degrees in both mechanical and electrical engineering from the University of Idaho. He started his career as test engineer with the General Electric Co., and then became instructor and later assistant professor of mechanical engineering at the University of Idaho for 11 years. He is the author of papers on metal forming, and articles on "Stresses in Concrete Induced by Various Temperatures," among others. THOMAS H. HAZLETT, also senior research engineer at Lockheed, began his career at Lockheed in stress work, later entering the field of production research, specializing in hydro-press forming and deep drawing. A graduate of Allegheny College, Mr. Hazlett has published articles on "Forming Contoured Sheet Metal Parts by Stretching," and "Comparative Formability of Various Aluminum Alloys."

THE modern airplane is constructed largely from sheet metal. As such, the most important production problems are those of sheet metal forming and assembling. Production is here considered as not only the act of forming and assembling the required number of parts, but also the making of forming tools, and all processing of parts such as heat-treating. Only that phase of the above concept of production which deals with the tooling for production and the forming and heat-treating will be considered here. The design of the aircraft parts will also be discussed somewhat, for it is obvious that the design of the part (designed shape and materials used) frequently determines whether the part can or cannot be readily made.

The great expansion in the aircraft industry brought on by the war has introduced many novel problems. Where formerly the quantities of a certain model were 10 or 20, the quantities now often range into the thousands. Furthermore, the war has emphasized the need for speed in production to an extreme emergency level.

In addition to this, the rapid advances in aircraft design make it necessary to incorporate design improvements in the form of frequent design changes in existing models; and to replace obsolete models with new ones. In order to meet these requirements, the production method must be maintained flexible; and technical knowledge must be available in advance, whereby forming tools can be quickly and accurately designed. As an example, Lockheed was recently requested to add a major item of equipment to a current model. This item was to be made from sheet metal and required the design and construction of about 10 dies. From the date of the request to the completion of the first unit required only five days, and at the end of two weeks the item was in full production.

The first and foremost problem in aircraft production is one of proper design and speedy production of the forming tools in the required quantities. In order to solve the problem, it becomes of prime importance to develop the method of die design and construction to such a degree that little of the expensive and time-consuming trial-and-error development is needed. This involves consideration of materials best suited to the conditions for die making, methods of die making, and determination of the forming limits of the materials to be formed (the extent to which a material may be deformed in forming a part).

The second problem is one of design of the aircraft part. Some ways in which aircraft design can assist in the production are as follows: Materials may be selected for formability where the strength considerations permit. (In general, the stronger the material, the lower is the formability.) While it is recognized that in many cases the strength and weight of a material must be given first consideration, it remains that there are many parts in which stresses are sufficiently low so that some thought can be given to selecting a material for formability. In parts that distort badly during heat-treatment, an advantage would be gained by selecting a material which need not be heat-treated. It is also quite obvious that there are a number of ways of designing for a given function. Of the possible designs, some will be found that are more readily formed than others. If the limits to which parts may be formed by the common methods of forming are known, a very large percentage of all parts can be designed so that they may be formed in one or a reasonable number of operations.

[This paper was presented at the SAE National Aircraft Production Meeting, Los Angeles, Calif., Oct. 1, 1942.]

in HIGH-PRODUCTION SHEET METAL FORMING

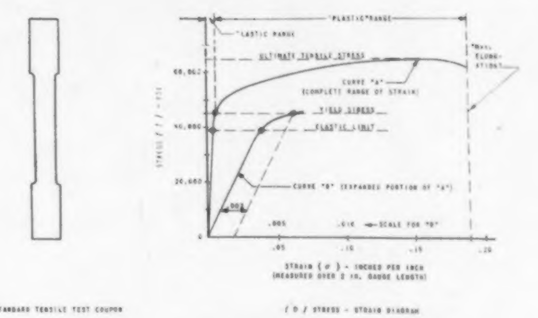
Without such knowledge, or application of such knowledge, many parts are unnecessarily difficult, if not impractical, to form. If the impractical design is not recognized, dies are usually constructed with the result that money and precious time and material are wasted before the part is redesigned. In such cases, it would have been much better if such trouble could have been anticipated and avoided. It is reasonable to assume that when designs are made with a consideration of the forming limits of the material, the problems in production will be greatly simplified and many will be eliminated.

With this in mind, research carried on at Lockheed has been directed first and foremost toward the analysis of basic forming operations and determination of forming limits. The method has been, on the one hand, to work with the production departments in the matter of "tool try-out" and to conduct numerous experimental investigations; and on the other hand, to study the forming operations analytically in terms of the plastic properties of materials to show the relation between cause and effect in forming. Typical subjects which have been investigated are:

1. Methods of measuring strains, and basic laws of deformation[1].
2. Properties of the structural materials (with regard to forming).
3. Spring-back and work-hardening properties of materials.
4. Materials best suited to the production of dies for aircraft production[2].
5. Coefficient of friction of dry and lubricated surfaces.
6. Forming limits in bending, stretching, drawing, and various other forming operations.
7. Lubricants for forming.

■ Properties of Common Aircraft Materials

Any discussion about sheet metal forming must of necessity hinge on the properties of the materials in use. The properties of a material are most commonly judged by the yield strength, ultimate strength, elongation in 2 in., and in some cases reduction of area at the break. These measurements are made from a standard tensile coupon, Fig. 1, which is pulled to failure. Reduction of area is usually only determined when round test specimens are used.



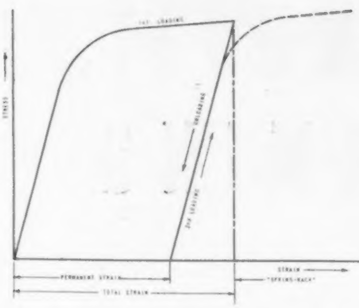
■ Fig. 1 – Typical stress-strain diagram based on tensile test

While the properties mentioned above are usually sufficient to judge the structural value of a material, such as the strength and ability to withstand static and dynamic loads, they do not necessarily give enough information to judge the forming characteristics and to analyze typical forming operations. Of more universal value is the stress-strain curve derived from a test of the tensile coupon. Since a detailed discussion of the method of applying stress-strain diagrams in the analysis of sheet metal forming operations has been published recently,[3] only a brief discussion will be given here. Fig. 1 shows a standard tensile coupon and a stress-strain diagram which describes graphically what happens when the tensile coupon is pulled to failure. The value of the stress-strain diagram lies in the fact that it shows the stress and strain relation during plastic deformation. For structural design purposes, it is the first part of the curve up to the yield strength that is of particular interest, while in forming, it is the portion of the curve from the yield stress to failure that must be considered. As the tensile coupon is loaded, it elongates, first of all, elastically, then plastically. Within the plastic range, the strain under load is part elastic and part plastic; that is, if the coupon is unloaded it contracts lengthwise, but not to its original length, Fig. 2. The tendency to contract upon unloading is termed elastic recovery, and is the cause of the spring-back phenomenon. Up to the so-called ultimate strength, the strains may be assumed to be uniform over the length of the test coupon between shoulders. When the ultimate strength is reached in a ductile material (the types used for forming aircraft parts), a local weakness results, in the form of a Lueders' line in some cases, or a general constriction (neck) in other cases. During the stage of necking, the material no longer deforms uniformly along the entire length, but only locally in the necked region. A typical strain distribution in the coupon after failure of the coupon is shown in Fig. 3.

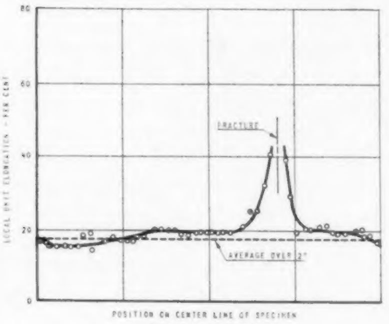
[1] See the *Journal of Aeronautical Sciences*, Vol. 9, No. 1, Nov., 1941, pp. 1-7: "Determination of Strain Distribution by the Photo-Grid Process," by G. A. Brewer and R. B. Glassco.

[2] See *Iron Age*, Vol. 149, No. 22, May 28, 1942, pp. 37-41: "Drawing Dies for Airframe Stampings," by G. A. Brewer.

[3] See the *Journal of the Aeronautical Sciences*, Vol. 9, No. 9, July, 1942, pp. 313-333: "Elastic Theory in Sheet Metal Forming Problems," by F. R. Shanley.



■ Fig. 2 – Stress-strain diagram for successive loadings



■ Fig. 3 – Distribution of elongation – measured over 0.02-in. gage lengths in a standard tensile test specimen of 24S-T alclad

An examination of this figure makes it clear that the elongation in 2 in. is an average value of strain which is essentially non-uniform. It is also clear that if elongation were measured on a gage other than 2 in., the value would be different than for 2 in. In most of the aluminum alloys, the maximum strain, measured in a short gage length, is much greater than the 2-in. elongation; or putting it in another way, the material is capable of much greater local deformations than the 2-in. elongation measurement indicates. This is very significant in some forming operations, and for this reason it has been found valuable to determine the elongation in a very short gage length, as 0.01 in. or 0.1 in. in the vicinity of the break. (This is a measurement quite analogous in its significance to the reduction of area commonly determined for round specimens.)

The most common and widely used sheet metals and the ones that will be discussed are the aluminum alloys 3S, 24S, 52S, 53S, and 61S. While it is realized that there are other aluminum and magnesium alloys and some steels which may come into wide use, this paper will be restricted to the above most widely used materials. In the designation of an aluminum alloy, as 24S, the number stands for the chemical composition, while the letter S means that it is a wrought material. The approximate chemical compositions of the common aluminum alloys are shown in Table 1. These alloys may be further classified as "heat-treatable" and "strain-hardening" alloys. (Of the alloys mentioned, 24S, 53S, and 61S are heat-treatable and 3S and 52S are "strain-hardening.") It is not the purpose of this paper to discuss the metallurgy of these alloys in detail, but merely to define the terms that will be used. A "heat-treatable" alloy is one which can be appreciably strengthened by heat-treating and quenching in some prescribed manner, while a "strain-hardening" alloy is one which cannot be appreciably strengthened by heat-treatment. While *all* wrought materials strain-harden to a greater or lesser degree, the "strain-hardening" alloys are those which can be strengthened only by cold working.

Table 1 – Chemical Composition of Some of the Common Aluminum Alloys*

	Alloying Elements, %				
Alloy	Copper	Silicon	Manganese	Magnesium	Chromium
3S	...	...	1.2	...	...
24S	4.5	...	0.6	1.5	...
52S	...	...	...	2.5	0.25
53S	...	0.7	...	1.3	0.25
61S	0.25	0.6	...	1.0	0.25

* See Alcoa Aluminum and Its Alloys, p. 84, 1940, Aluminum Co. of America, Pittsburgh, Pa.

The wrought aluminum alloys in their annealed state are designated by adding the letter O, as for example 24S-O, which means the annealed state of 24S. The heat-treated state of those which respond to heat-treatment is indicated by the letter T, as in 61S-T and 24S-T. The heat-treatment consists of heating to a prescribed temperature and quenching or cooling at a relatively high rate. In general, however, the heat-treated material does not attain its maximum strength immediately after the quench, but remains soft and formable for a period of time. This state is designated by the letter W, as for example 24S-W, 53S-W, and 61S-W. For some alloys, such as 24S, the heat-treated material hardens or "ages" rapidly and attains most of its full heat-treated strength within several hours after the treatment, while for other alloys, such as 53S and 61S, the rate of aging at ordinary air temperatures is so slow that the material does not attain its S-T strength within a reasonable length of time. The latter materials are converted into the full S-T condition by heat-treating or "artificially aging" at a moderate temperature, 300 to 400F. Typical mechanical properties of the common "heat-treatable" alloys in the three states are shown in Table 2.

Sometimes the strength of heat-treated material is further increased by cold rolling. Such material is then designated by S-RT, as in the case of 24S-RT.

Such "heat-treatable" materials as 53S and 61S may be used in the airplane structure in either the S-O, S-W, or S-T states. In the case of 24S, the corrosion resistance is insufficient in the S-O state; as a result, this material is always used in the S-T or heat-treated state in the structure even though the added strength is not needed. In the past, it has been customary to form the material *before* heat-treatment; that is, in the S-O state; but the new techniques and knowledge are making it increasingly possible to form many parts directly in the S-T condition, thus eliminating troubles due to distortion during heat-treatment.

Work-hardened tempers of the "work-hardening" alloys, like 52S, are usually designated as ¼H, ½H, ¾H, or H, depending upon the degree of cold working. Such material may be restored to the annealed or S-O state by heat-treatment. Typical mechanical properties of the work-hardening alloys are also shown in Table 2.

Basic Forming Operations

A study of the forming operations encountered in aircraft production has led to grouping all operations into several basic groups. These are bending, stretching, shrinking, and drawing.

Table 2 – **Mechanical Properties of Common Aluminum Alloys with Various Tempers***

Alloy and Temper	Tension Properties: Yield Strength, psi	Tension Properties: Ultimate Strength, psi	Elongation % in 2 in.
3S-O	6,000	16,000	30
3S-½H	18,000	21,000	8
3S-H	25,000	29,000	4
24S-O	10,000	26,000	20
24S-W Alclad	......	......	..
24S-T Alclad	41,000	62,000	18
24S-RT Alclad	50,000	66,000	11
52S-O	14,000	29,000	25
52S-½H	29,000	37,000	10
53S-O	7,000	16,000	25
53S-W	20,000	33,000	22
53S-T	33,000	39,000	14
61S-O	8,000	18,000	22
61S-W	21,000	35,000	22
61S-T	39,000	45,000	12

* See Alcoa Aluminum and Its Alloys, p. 89, 1940, Aluminum Co. of America, Pittsburgh, Pa.

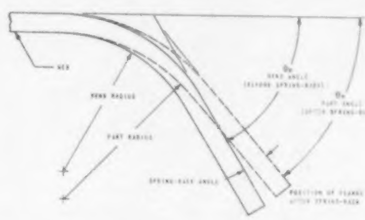
Before going into a discussion of forming limits, it is desirable to discuss these simple basic operations in some general detail and to give simple examples of each.

Bending – Bending is actually a combination consisting partly of stretching and partly of shrinking, two operations which are discussed later in this paper. However, due to peculiarities of its own, it is desirable to consider it as a basic forming operation. A cross-section through a part that has been bent in a simple manner on the power brake is shown in Fig. 4. In the case of the part shown in the

■ Fig. 4 – Cross-sections of typical sheet metal bends

figure, the indentations in the surface were uniformly spaced in the flat sheet. The difference produced in the spacing by forming clearly shows the strains that result from bending. Bending is characterized by the fact that the fibers on one surface become elongated, while the fibers on the other surface are shortened. Between the two surfaces the strain varies from compression on one side to tension on the other. On one intermediate surface, called the neutral surface of bending, the strain is zero.

Another characteristic of bending is angular spring-back. A part bent to an angle θ_b, Fig. 5, returns to some angle θ_p when the forming forces are removed. The difference between the angle of bend and final-part angle is the "spring-back angle." Similarly, there is an increase in the radius of the bend.



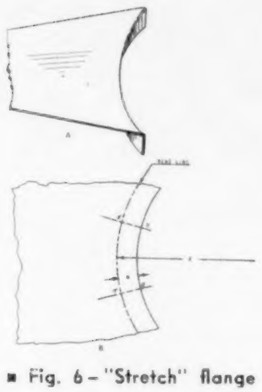
■ Fig. 5 – Effect of spring-back on bend

While bending may be considered as the simplest forming operation to perform, difficulties arise in the high-production methods due to spring-back. In order to produce a bend with a desired angle it is, therefore, necessary to anticipate the value of the spring-back angle, and overform by this amount; that is, if dies are constructed for bending, it is necessary to make spring-back allowance. In some cases, especially when the radius of the bend is relatively large, it is also necessary to make allowance for the change in radius which results from elastic recovery, by making the die, or block radius, smaller than the desired part radius. When the desired part radius is itself quite small, it is usually possible to make parts to the prescribed tolerance without this refinement.

The amount of elastic recovery or spring-back varies approximately in proportion to the yield strength of the material,[3] assuming other factors to be the same. An examination of Table 2 reveals that, on this basis, 24S-O has only about one-fourth the spring-back of 24S-T. Elastic recovery is also the cause of much "canning" (the type of canning which results from forming). In the final analysis, it is the *small amount of elastic recovery* of 24S-O which constitutes one of the main arguments in favor of forming parts from 24S-O and heat-treating the parts, rather than forming the part from 24S-T directly.

This advantage of forming from 24S-O material and heat-treating is frequently offset by distortion arising from heat-treatment, the cost of additional operations, and distortion removal. In some cases, spring-back can be eliminated by a subsequent or simultaneous stretching operation[3] (see next section).

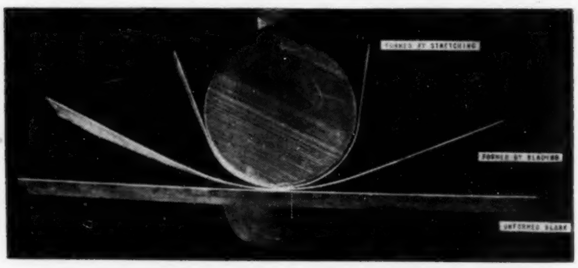
Stretching – A typical stretching operation results in forming a concave or "stretch" flange (see Fig. 6). In forming this part from the blank shown in Fig. 6(B), the point a is moved to a point directly under a', and b to a point under b'. Thus the length ab is increased or stretched to the length $a'b'$. Actually, there are few if any forming operations which are pure stretching; instead, most stretching operations are associated with some simultaneous bending, as in the case cited. An operation is here considered as stretching when the predominant strains are due to stretching rather than to other effects, like bending. The strains due to stretching are all tensile strains, while bend-



■ Fig. 6 – "Stretch" flange

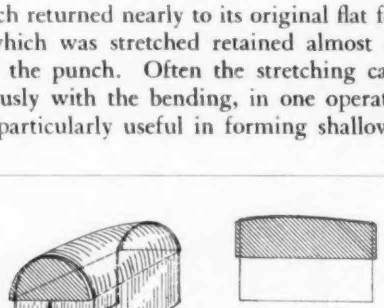
ing produces both tensile and compressive strain in neighboring layers of the sheet.

Although the permissible strains are low when forming by stretching, a valuable characteristic is that elastic recovery does not cause appreciable change in radius or angle of a curved part. For this reason, it is sometimes advantageous to follow a bending operation with a stretching operation to remove angular spring-back. This is illustrated in Fig. 7. One part was bent to the contour of the punch, and the other was bent and then stretched over the die. The part which was merely bent to the contour



■ **Fig. 7 – Comparison of angular spring-back for bending and stretching – view showing the unformed blank, the partially formed piece (formed only by bending) (large spring-back), and the fully formed piece (formed by bending and stretching) (minimum spring-back)**

of the punch returned nearly to its original flat form, while the part which was stretched retained almost exactly the contour of the punch. Often the stretching can be done simultaneously with the bending, in one operation. This process is particularly useful in forming shallow-curvature

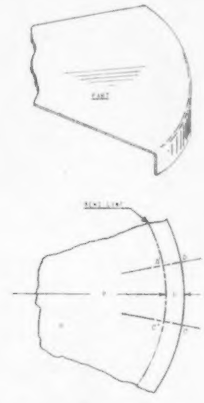


■ **Fig. 8 – Stretching to a doubly curved surface**

parts such as skins (see Fig. 8), which will be discussed later.

In most cases of stretching, special grips must be employed to apply the large stretching forces. Such grips are a subject for careful design, since they must apply tensile forces nearly equal to the ultimate strength of the material being formed without damaging the strength. Even slight damage of the sheet usually results in rupture at greatly reduced values of strain.[3]

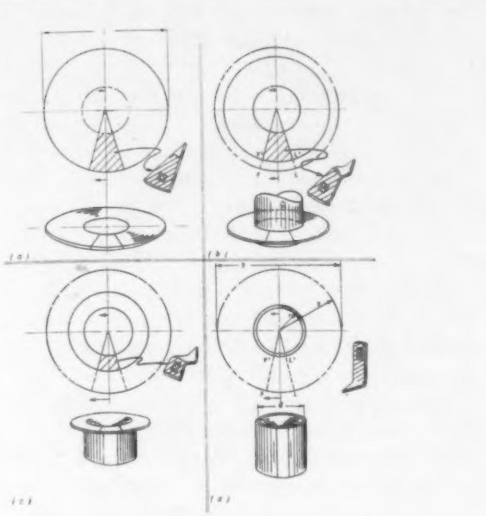
Shrinking – Shrinking is a forming operation theoretically just the negative of stretching. A simple case of shrinking as a basic forming operation is illustrated by the "shrink" flange, shown in Fig. 9. In this case, the



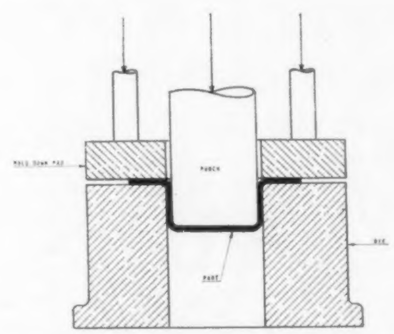
■ **Fig. 9 – "Shrink" flange**

length *ab* must be shortened to the length *a'b'*. However, any similarity between simple stretching and shrinking ends at this point, due to the practical fact that in order to do any appreciable shrinking of a sheet material, it is necessary to give support to the material against buckling or wrinkling. With proper support against buckling, it is possible to produce very large compressive deformations in a material without danger of producing failure. However, the actual mechanics of providing support against buckling during forming is not a simple matter and only a very few recognized practical means have been developed. Without the support to buckling, only slight amounts of compression can usually be produced in sheet metal.

Drawing (deformation under combined tension and compression) – The majority of forming operations are not strictly stretching or shrinking but a combination of the two. When stretching takes place in one direction and shrinking takes place simultaneously in the normal direction, the forming operation will be called drawing. A typical drawing operation is shown in Fig. 10, in which a flat round blank is converted into a cup-shaped part. In this method of forming, a sector of the circular blank becomes an axial strip in the side of the cup. Hence, a small element in the blank (shown shaded) becomes elongated in one direction and compressed in a direction normal to the former. It is this particular combination of tensile and compressive stress which characterizes the drawing operation. Due to the usual tendency to buckle under the compressive stress, it is necessary to give support to the material during forming. One typical form of die for drawing is shown in Fig. 11. It consists of a punch, die, and hold-down pad. The die as a whole is usually mounted in a so-called double-acting press, which is so arranged that the punch or hold-down pad can be moved



■ Fig. 10 – Four stages in a cup drawing operation



■ Fig. 11 – Typical draw die arrangement

each independently of the other or in any desired sequence. In operation, both punch and hold-down pad are raised, and the blank is placed in position. The hold-down pad descends first and applies pressure on the blank, which prevents buckling as the punch descends and draws the blank into the die cavity.

The tension which results in drawing may be considered entirely due to the resistance of the segment of the blank to being drawn into the cup and not due to the gripping action of the hold-down. Only enough hold-down pressure is usually applied in this case to prevent wrinkling. The compression referred to results from the reduction in radius of all the circular elements originally in the blank to smaller circular elements in the cup.

At the bottom of the cup, the material is subject to a tensile load by the punch tending to push the bottom out. At the extreme rim, the material is subjected to compression in a circumferential direction, while at intermediate radii the combined tensile and compressive stresses exist.

Under the action of combined tension and compression, it is possible to produce large deformations without causing fracture of the metal. Simultaneous strains of 70% in tension and 50% in compression have been observed in a cup drawn from a material with a typical elongation at failure of 20% in tension.

Combinations of basic forming operations – In addition to the basic operations mentioned, operations which are a combination of these basic operations may be recognized, as for example bending and stretching, bending and drawing, and stretching and drawing. The former two terms are self-explanatory and the latter differs from drawing in that additional hold-down pressure may be applied to produce a partial gripping action between blank and hold-down surface, and between blank and draw ring. In this case drawing, with the stretching action accentuated, results.

■ Classification of Actual Parts

The basic forming operations were described in terms of simple elementary parts. Actual parts have an infinite variety, and it is not usually possible to say that a specific part is this or that simple type. It is also true that the desired shape of a part does not limit the method of forming to one particular basic method. Usually a part can be formed in several different manners, of which one is better adapted than the rest. The success of most forming work depends upon selecting the best adapted method and upon the skill and technique of the tool designer, tool maker, and press man. The proper selection of a forming method must be based upon an understanding of the basic forming operations and a knowledge of the forming limits for each. The classification of parts will be discussed first. Forming limits and techniques will be taken up in another part of the paper.

On the basis of data obtained from a large number of tool tryouts, it has been found possible to classify the great variety of parts into four groups and to arrive at some useful conclusions. In this discussion, it must be borne in mind that the consideration of the manner of forming a part must depend upon the shape of the part as it will emerge from the forming die before trimming to final form. Any excess material that must be added to the blank must be considered as a portion of the part during forming. It also happens in some instances that it is easier to form a shape which can be cut into a number of desired parts. Obviously, in this case the part which is formed does not necessarily resemble the desired part. The classification of shapes (as formed) is as follows (see Fig. 12):

1. Parts with straight flanges.
2. Parts with curved flanges.
3. Drawn parts, as cups and boxes.
4. Smoothly contoured parts, as "fairings" and skin surfaces.

Each of the classifications will be discussed briefly.

■ Fig. 12 – Typical part of each class

Parts with straight flanges – This class of parts, Fig. 12(*A*), is nearly always formed by bending. The forming limits are usually expressed by tables of design values of bend radii. In this class of parts, angular spring-back is quite large and "spring-back allowance" must be made. Parts with slight curvature in the mold line may be included in this class and can be formed in the same general manner from such materials as 3S-O, 52S-O, and 24S-O. Such parts can usually not be formed from materials with high yield strength, as 24S-T.

Parts with curved flanges – Forming parts with curved flanges, Fig. 12(*B*), involves shrinking or stretching, in addition to bending found in straight flanges. The forming limits consist of (1) minimum-bend radii, which apply to bending of flanges as in the first class, and (2) permissible tensile or compressive strains in the edge of the flange. The permissible forming tensile strain in a stretch flange is relatively low, and will be discussed in more detail below. Determination of permissible strains requires, first, that a method for predicting forming strains be developed, and second that the maximum permissible strains be determined. The method has been to measure strain in typical forming operations and correlate these measured strains with the geometry of the part in the form of an equation for strain. For example, for stretch flanges it has been found in this manner that the tensile strain in flanges may be expressed by the equation[3]:

$$e = \frac{w}{R} \qquad (1)$$

where:

e = the critical strain, that is, in the fiber at the blank edge

w = the width of the flange (see Fig. 6)

R = the radius of curvature of the bend mold line

Forming limits are then determined by forming parts with various values of w and R until values are found that just do not cause failure, and substituting these values into equation (1). In design, the same equation is used to predict the strain for a proposed design. If the value of strain is less than the permissible value, the design may be considered safe.

Permissible strains for shrink in flanges which are unsupported against buckling are very low. For this reason, most flanges of this type are formed by a method such as drawing, which supports the material against buckling. Some novel methods, other than drawing, for forming shrink flanges have been developed and will be discussed later.

Drawn parts – This class comprises such parts as cup- or box-like shapes, with or without flanges, Fig. 12(*D*). Such parts are formed with dies similar to that shown in Fig. 11 for a simple cup. As has already been described, the forming stresses are combined tension with compression (the two stresses being perpendicular). Due to the combined stress, very large forming strains are permissible. With proper design of such dies, it is also possible to form very intricate parts by this method. Parts too difficult to form in one single operation can frequently be made in several stages.

Such aluminum alloys as 3S-O, 52S-O, 52S-½H, 53S-O, 53S-W, 61S-O, 61S-W, and 24S-O are highly formable in drawing (the actual order will be discussed later) while the heat-treated and aged alloys like 24S-T are relatively less formable.

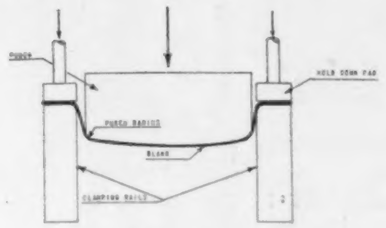
While the forming limits for drawing may be indicated in terms of permissible strains, this is not very practical for the reason that it is not a simple matter to predict the strain. Since drawing is associated with parts more or less circular in outline, a more convenient way to express the forming limits is in terms of the radius of the curved outline of the part and the corresponding radius of the blank. For the cup shown in Fig. 10, the forming limit is usually determined by drawing cups with successively larger blanks until the maximum size that can be drawn into a cup without failure is reached. Failure may result from tearing out the bottom, cracking in the rim, or surface cracking in the sides of the cup. (The latter is sometimes found in first-operation draws, but usually occurs only in the later stages of a multiple drawing operation). The forming limit is expressed as the ratio of the radius of the cup divided by the maximum blank radius. This is the minimum $\frac{r}{R}$ which may be formed, where r is the radius of the cup and R is the maximum radius of the blank. The forming limit for cups may also be expressed as $\frac{h}{r}$, where h is the maximum height of cup which may be successfully formed.

In other drawn parts, such as boxes, one can analyze the part as consisting of portions of boxes connected together with flat sections of material. For example, a rectangular box may be thought of as consisting of four quarter cups for corners connected with flat side panels. Theoretically, the corners require drawing while the sides do not. Practically the drawing action (forming with combined tension and compression) cannot be confined only to the corners, but the whole part is formed by a modified drawing operation.

Smoothly contoured parts – A typical smoothly contoured part is shown in Fig. 12(*C*). Skin panels, fairing, and related parts fall into this class. Such parts can sometimes be drawn in multiple-action presses; however, in many cases it is very difficult if not impossible to support the material against buckling in drawing. Many such parts can best be formed by stretching, or by a combination of stretching and drawing. As already stated, the distinction between stretching and combined stretching and drawing is not necessarily great. In some cases the die arrangement for stretching is different from the arrangement for drawing, while in other cases there is no difference. In the case when a draw die is used, the difference between drawing and stretching is that for drawing the hold-down pressure is so regulated that material slips under the hold-down pad into the die cavity, while for stretching the hold-down pressure is increased to such an extent that little or no slipping takes place under the draw ring, and forming takes place by stretching.

In many cases, a modified die arrangement, simpler than for drawing, is utilized. A typical arrangement of a punch with two rails and hold-down pads is shown in Fig. 13. By this method, the blank is clamped on two sides while the punch is pushed against the sheet.

Parts which overlap the above classification consist of



■ Fig. 13 – Stretch die arrangement

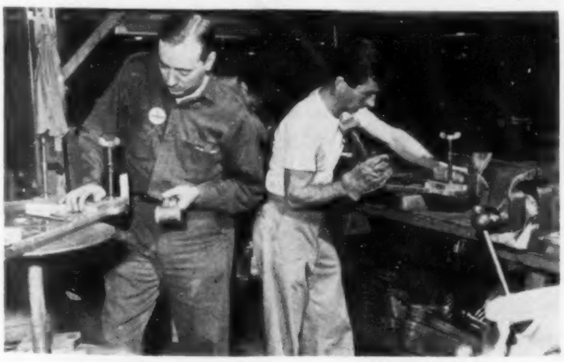
parts which fit into more than one classification. A portion of the part may have straight flanges, while another part has curved flanges, and a third portion is smoothly contoured. In general, if the part is such that it can be formed, the forming limits for bending apply to the portion to be bent, the forming limits for stretching apply to the portion to be stretched, and so on.

■ Forming Equipment

The various types of equipment used to produce the fundamental operations and types of parts just discussed will now be reviewed in the light of the functions performed by each. Some of the methods about to be mentioned are rapidly becoming obsolete within the industry; however, they are all utilized to a greater or lesser degree at the present time.

The most elementary of all forming techniques is that of contouring a piece of sheet metal by hand-forming with a hammer. Although this process has been virtually eliminated from the modern production line in the sense that parts are formed completely by this method, it is still maintained as a method of correcting the deficiencies produced by other types of equipment.

A view of a portion of a hand-forming section is shown in Fig. 14. Two main classes of parts are being routed through this section: (1) parts incompletely formed on some other piece of equipment, usually the single-acting press, which utilizes a confined rubber punch, and (2)

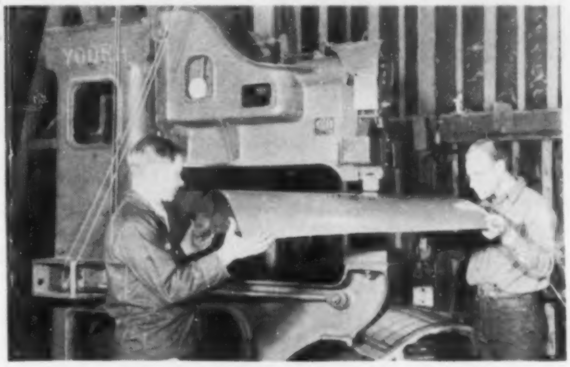


■ Fig. 14 – Hand-forming with rawhide mallets

parts which were initially formed in the "S-O" or soft temper condition and subsequently were heat-treated with the attendant warpage and distortion.

The first class consists primarily of parts containing severe convex or "shrink" flanges, or joggles which, due to design limitations, cannot be completely formed by one of the devices which will be described later. Minor distortion of certain types of parts formed directly in the "S-T" or heat-treated temper may also be corrected by hand-forming. The second class is made up largely of parts having rather large flat areas, which are free to distort due to the rapid quench in heat-treating.

A modification of hand-forming with a mallet is the use of a power driven planishing hammer, such as is shown in Fig. 15. Comparatively large local forces can be exerted



■ Fig. 15 – Planishing hammer

by these machines, which are used primarily for removing very tight wrinkles developed during forming. They are also utilized to a limited extent to make slight changes of contour in parts.

Although not strictly a method of hand-forming, shaping of shallow skins by crown rolling, as shown in Fig. 16, is closely analogous. This operation may follow partial form-

■ Fig. 16 – Crown rolling shallow-contoured skins

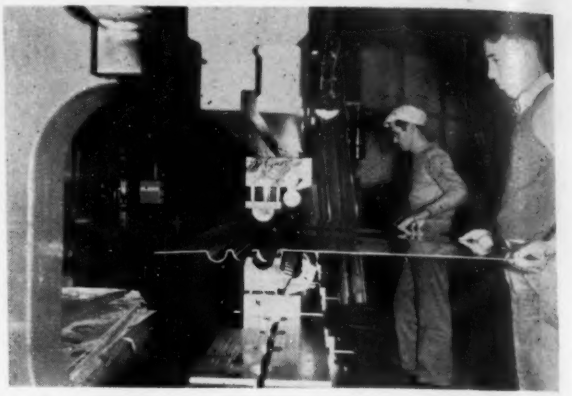
ing on drop hammers, rubber press, or, in the case of very shallow contours, may be the only forming operation performed.

The same basic shortcomings apply to all these methods of forming, namely: (1) low production rate, (2) high unit cost when produced in quantity, (3) lack of uniformity, hence lack of interchangeability, and (4) limited degree of deformation of the metal.

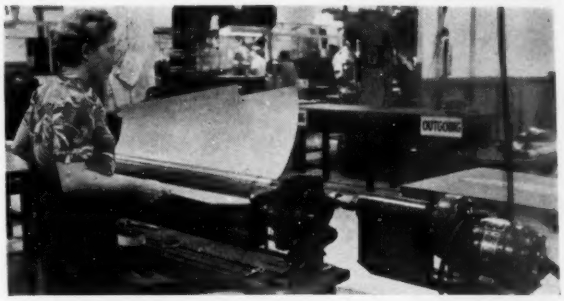
Drop hammers (see Fig. 17) were adopted within the industry in the era of low production as a means of obviating the fourth objection just mentioned, namely, the limited degree of formability obtainable by the other methods. A simple two-piece mating die is used, usually consisting of a zinc-alloy die and a lead punch which has been cast directly into the die.

Successful employment of this equipment requires a high

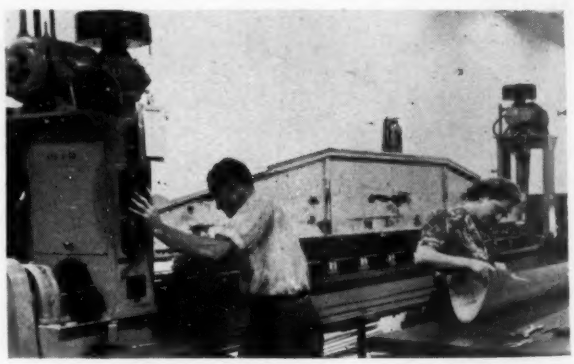
■ Fig. 17 – Forming a severely contoured part on the drop hammer



■ Fig. 18 – Compound die for power brake



■ Fig. 19 – Small power rolls for forming single-curvature parts



■ Fig. 20 – Rolls for making single-curvature parts with varying radii of curvature

degree of skill and many practical devices are used to increase its usefulness. Among these might be mentioned the judicious use of rubber and plywood rings, of which the latter aids particularly in forming deep sections by partially supporting the outstanding flange against wrinkling. However, without some device which will exert constant pressure against the blank during the time it is being drawn into the die, considerable wrinkling will result.[4] Since careful control of wrinkling is necessary to produce a good part, considerable hand work is necessary in drop hammer forming, as shown in Fig. 17. In some cases, the part may be partially formed, wrinkles removed by the planishing hammers, and the part returned to the drop hammers for final shaping. Such time-consuming operations result in a low production rate for deep sections.

However, these hammers have their place even today and are quite efficient in forming certain parts which do not involve a large amount of compressive strains in the metal. They are particularly well adapted for removing distortion from parts which were formed on either the rubber press or double-acting press.

The problem of uniformity has not been eliminated completely but has been considerably decreased in the past few years with the introduction of a zinc-aluminum alloy which retains its shape quite well after repeated impact loads. This is marketed under the trade name of "Kirksite."

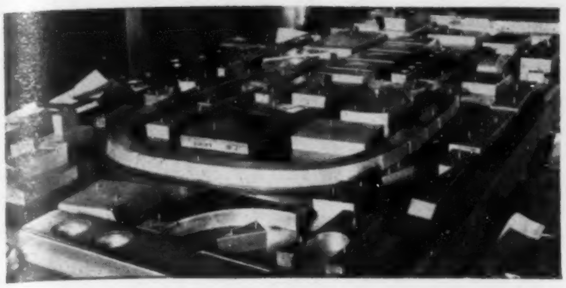
Probably the simplest type of equipment which permits relative precision in forming is the power brake. Although its use is limited to single-curvature parts, applications of it are increased considerably within the aircraft industry when adapted with compound dies such as the one shown in Fig. 18. The die shown is used for making corrugation panels and has been found quite efficient in this application.

Single-curvature parts having a rather large radius are simply and quickly formed on the small power-driven rolls, shown in Fig. 19. The radius of curvature may be changed at any point in the forming operation either by hand or by a suitable cam arrangement which makes the machine entirely automatic. Parts having a single curvature, but with different radii of curvature on each end, may be formed on the rolls shown in Fig. 20. With this machine, the radius on either end of the part may be varied independently as long as the two ends can be connected by a straight-line element.

Although the common mechanical punch press is of unquestionable value in any general sheet metal shop, its use within the aircraft industry is limited. This is primarily due to the wide use of large single-acting presses which use a confined rubber punch and simple form block. Punch presses, however, do play an important part in the fabrication of small parts, particularly those having a rather severe shrink flange. Small drawn shapes are also made on these presses by using double-acting dies with spring- or air-actuated hold-downs.

The majority of rather simple flanged parts are made on a large single-acting hydraulic press which utilizes a confined rubber blanket in the upper platen and a simple male form block on the lower table. A view of a typical loading arrangement for such a press is shown in Fig. 21.

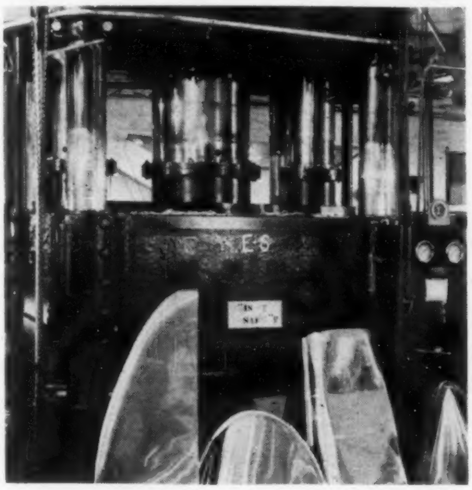
[4] See *Aero Digest*, Vol. 40, No. 2, February, 1942, pp. 126-135: "The Mechanics of Deep Drawing Sheet Metal Parts," by G. A. Brewer and M. M. Rockwell.



■ **Fig. 21 – Table loading arrangement for rubber pressure press**

The economic advantage of this process is evident when the relative cost of punch press dies is compared with that of simple form blocks. The latter do not require careful mating of component parts. The usefulness of this process is further emphasized when it is realized that 45,000 parts have been made on such a press in a single day. Parts containing simple flanges, joggles, beads, and flanged lightening holes are best suited for this type of forming. The forming limits and techniques will be discussed more fully in another section of this report.

A slightly different application of a rubber punch is shown in Fig. 22. In this case, skins and fairings are formed in a rather large Kirksite die. Usually, rubber

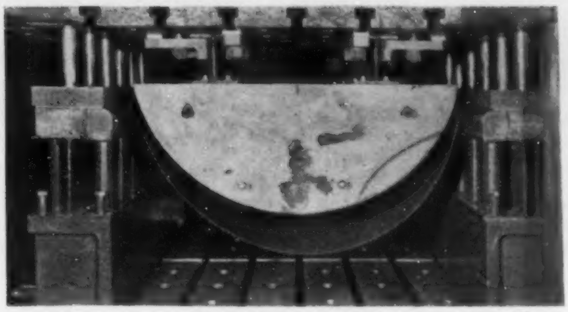


■ **Fig. 22 – Rubber pressure equipment used to form skins and fairings**

blankets are so placed that the sheet metal blank is partially locked along the sides of the die, resulting in a stretching of the sheet. In general, parts made by this process are formed in the "S-O" condition and subsequently must be heat-treated, with considerable distortion resulting.

To overcome this objection, which is particularly serious in the case of parts with rather shallow contour, a method of stretch-forming parts directly from "S-T" material was developed. This process makes use of a large double-acting press set-up as shown in Fig. 23.

Supporting rails are placed at two sides of a punch which has been contoured to the desired shape. The blank is rigidly clamped in place between them by means of pressure plates actuated by the secondary ram of the press. The punch, which is attached to the main ram, is then



■ **Fig. 23 – Set-up for stretching skins in the double-acting press**

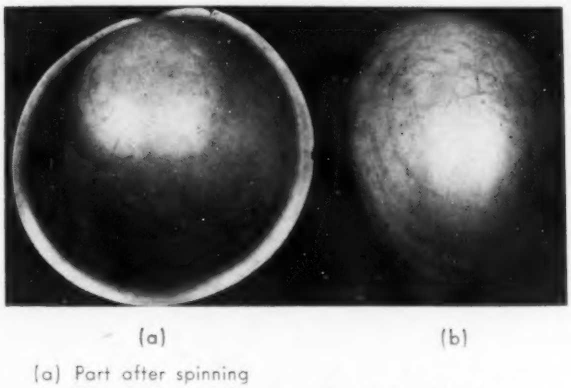
lowered until the sheet is stretched to conform to the contour of the punch.

It is quite possible to use a "stretching machine" in place of the double-acting press in stretch-forming operations. Such a machine has mechanically or hydraulically operated clamps which grip the sheet while it is being formed in a manner similar to that just described for the double-acting press. Most of these machines utilize comparatively low clamping pressures, and therefore have serrated grips to obtain more positive gripping action. However, tests at Lockheed indicate that a more favorable stress distribution is obtained inside the clamps by using smooth clamping surfaces and high normal pressures. In general, this has been found to eliminate tearing of the sheet within the grips.

Forming of this type is naturally limited to the tensile strains which the material will withstand. As a result, there are only two classes of contoured skins which can be formed in this manner: (1) parts having shallow contour in two directions, and (2) parts with severe contour in one direction and shallow in the opposite direction. Parts with severe contour in both directions must be formed in some other manner.

This latter class of parts may be formed by several of the methods already mentioned, but standard double-action draw dies are to be preferred when the depth of draw or wrinkling tendencies do not dictate an excessive number of redraws. In some cases, it has been found expedient to spin the part and give it a final shaping blow in the drop hammers. The part shown in Fig. 24 is an example of this.

The more important of the foregoing types of equipment will now be discussed in greater detail.



(a) (b)

(a) Part after spinning
(b) Shape of part after operation in the drop hammer

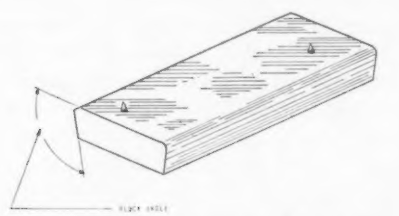
■ **Fig. 24 – Example of a part formed by spinning and shaping in the drop hammer**

Hydro-Press Forming with Rubber Platens

As mentioned previously, a large portion of aircraft parts are formed on a single-acting press using a confined rubber punch in the upper platen. This will be referred to as "rubber pressure forming" for the remainder of this paper. The type part best suited for this process, the limitations, and special aids to forming will now be considered in more detail.

Until the past two years, practically all forming was done on the S-O or soft temper materials. This was due to increased problems of forming the harder tempers, primarily caused by difficulty encountered from spring-back. As pointed out previously, this is a negligible quantity in the soft tempers but becomes a major factor in tool design when S-T is considered.

A series of tests was undertaken at Lockheed to evaluate the spring-back angle for 24S-T alclad parts having simple straight flanges.[5] Material thicknesses from 0.020 to 0.064 in. were formed over 1-in. high form blocks having external angles varying from 50 to 130 deg. A view of a typical test form block is shown in Fig. 25. The press was



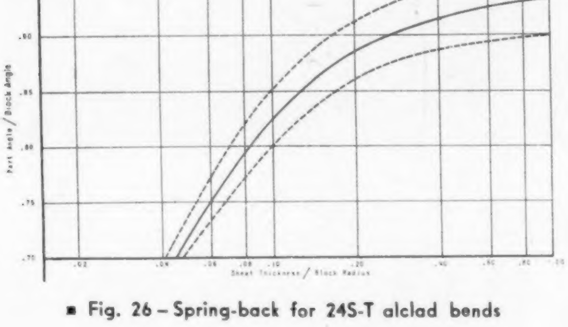
■ Fig. 25 – Typical form block used for tests

operated at a load calculated to produce an average pressure on the parts of approximately 1300 psi. The majority of tests were conducted with rubber having a Shore hardness of 60-65. A few parts were run with rubber of 70-72 Shore hardness. A production press was used under normal operating conditions.

Results of these tests are presented in the curve given in Fig. 26, which is a plot of two non-dimensional ratios.

[5] See *Product Engineering*, Vol. 13, No. 7, July, 1942, pp. 382-383: "Springback in Flanging," by F. B. Chapman, T. H. Hazlett, and William Schroeder.

[6] See *Iron Age*, Vol. 149, No. 25, June 18, 1942, pp. 41-46: "Hydro-Press Forming with Rubber Platens," by T. H. Hazlett and William Schroeder.



■ Fig. 26 – Spring-back for 24S-T alclad bends

It was found that when the ratio of (sheet thickness/block radius) was plotted against (part angle/block angle), virtually all points fall within the narrow band bounded by the two dotted curves shown. The central solid curve represents an average of these values.

Since a large proportion of production parts use a 90-deg flange, Table 3 has been prepared with the spring-back angles computed from the curve.

Table 3 – Spring-Back Angles For 90-deg Parts – 24S-T Alclad

	Material Thickness, in.					
Radius, in.	0.020	0.025	0.032	0.040	0.051	0.064
1/32	7¼	7	..	..	..	..
1/16	9½	8½	7¾	..	..	..
3/32	11¼	10	9	8¼	7½	..
1/8	13½	11½	10	9¼	8¼	..
5/32	16	13½	11½	10¼	9	8¼
3/16	18¼	15½	12¾	11	9¾	9
7/32	21	17½	14½	12¼	10½	9½
1/4	23¼	19¼	16	13½	11¼	10
9/32	26	21¼	17½	14¾	12¼	10½
5/16	28¼	23¼	19	16	13	11½
11/32	30¾	25½	20½	17	14	12
3/8	33	27¾	22	18¼	15	12¾
13/32	..	..	23¾	19½	16	13½
7/16	..	..		21	16¾	14¼
15/32	..	..		..	..	15
1/2	..	..	..	..	..	16

For part angles other than 90 deg, multiply spring-back angle by the following factors:

Part Angle	Factor	Part Angle	Factor
50	0.555	75	0.835
55	0.611	80	0.890
60	0.667	85	0.945
65	0.722	95	1.056
70	0.779	100	1.110

However, since production conditions rarely permit the close control necessary for all points to fall on the average curve, a tolerance of ± 2 deg should be permitted. This corresponds to the width of the band between the dotted curves, Fig. 26.

During the course of this investigation, it became clear that uncontrolled factors were influencing the results. Each possible variable was then carefully isolated and varied, one at a time.

The influences of each variable have been reported elsewhere[6] and are too complex to be repeated at this time. However, it should be noted that the magnitude of the unit pressure exerted on the part by the rubber punch, the proximity of adjacent objects or form blocks located on the table, rubber hardness, and form-block height all exert considerable influence on forming conditions. For this reason, it must be emphasized that the values given in Fig. 26 and Table 3 apply for the operating conditions stated, and a change of any one variable may affect the results. This is particularly important for the heavier gages.

The investigation of the individual variables showed a definite relationship between spring-back angle and pressure exerted on the form block. It was noted that, if the pressure *on the block* exceeded a certain critical value, the angle did not change with further increase in pressure. Thus it may be seen that spring-back angle may be stabilized considerably by operating at the maximum pressure practical for the equipment used.

The same investigation revealed that the unit pressure exerted on the form block may be materially increased by any of the following methods:

(1) Increase applied press load.

(2) Judicious use of "displacement blocks." These consist of metal blocks with a base 5 x 12 in. and 3½ in. high with well-rounded corners.

(3) Increase form-block height.

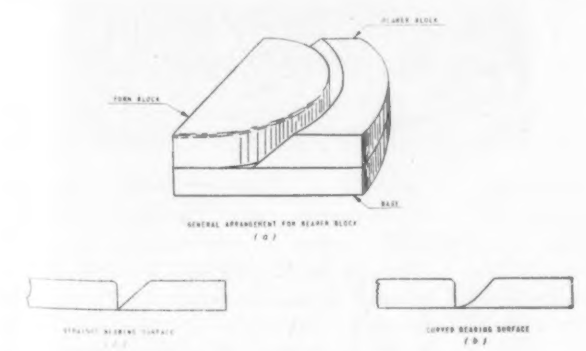
Care must be used when locating the number of displacement blocks on any one table loading. If an excessive number are used, the principal load will be taken by the displacement blocks, with the result that the form block will be subjected to *decreased*, rather than *increased* pressure.

The second major problem connected with forming parts by this process is the amount of curvature, both convex and concave, that is permitted in the mold line of a flanged part. The forming limit of parts having concave contours, such as the one shown in Fig. 6, is a function of the ductility of the material, since only stretching is involved. Tests at Lockheed indicate that parts may be designed with maximum tensile strains of 10-12% for 24S-O and 24S-T alclad material. Although much higher values have been obtained under certain conditions, this is considered the design limit for rubber pressure forming.

On the other hand, convex or "shrink" flanges, as illustrated in Fig. 9, are limited by the tendency of the material to wrinkle under compression. Since the rubber in the upper platen offers little restraint against wrinkling, the compressive strains in the material must be kept small. No single value for the critical strain can be given because the wrinkling is a function of the blank thickness and, to some extent, the material involved. Results of tests on 24S-O indicate that strains of 1-6% (depending on blank thickness) are possible without objectionable wrinkling. Using 0.040-in. thick blanks, a series of tests were run on 24S-O, 52S-O, 53S-W, and 61S-W to compare the shrink characteristics of these materials. No significant difference between them could be detected. Tests indicate it is not possible to obtain compressive strains larger than 0.5% for 24S-T without wrinkling.

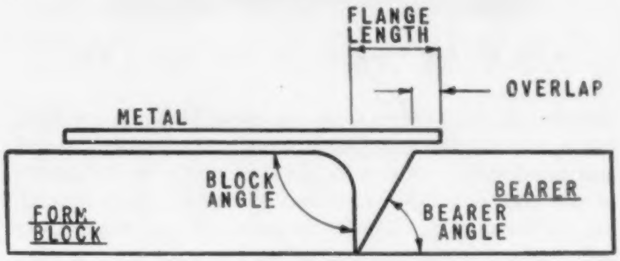
Since a large number of parts involve compressive strains greater than these values, a number of methods are used to increase the useful range. Until recently, the parts were permitted to wrinkle and the wrinkles removed by hand forming. Although this practice has been greatly reduced, it is still used in some cases.

Development of the so-called "bearer block" has increased the range of permissible strains considerably. This device consists of an auxiliary block placed adjacent to the regular form block in the region of the shrink flange, as shown in Fig. 27(a). The inner surface of the bearer block may be curved as in the cross-sectional view (b), or straight as in (c). The effectiveness of the two shapes is approximately equal, so the one with straight inner faces is to be preferred for the ease with which it can be constructed. Table 4 gives specifications for construction of such blocks. It should be pointed out that wrinkles will not necessarily be eliminated on the minimum-contour radius recommended in the table, but this device will prevent very tight wrinkles (which cannot be removed by hand forming) from occurring.



■ Fig. 27 – Bearer block

Bearer Blocks



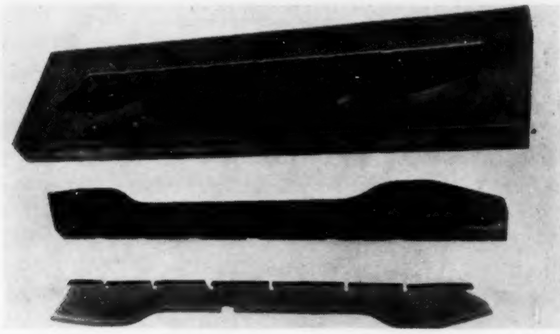
Material	Flange Length	Overlap Recommended	Minimum Flange Contour Radius Recommended for Bearer Application, In	Bearer Angle Recommended: 120-deg Block	90-deg Block	60-deg Block
0.020	½	1/16	3	65	55	45
	⅝	1/16	3	65	55	45
	¾	1/16	4	65	55	45
	1	1/16	5	65	55	45
0.026	½	1/32	3	65	55	45
	⅝	1/32	3	65	55	45
	¾	1/32	3	65	55	45
	1	1/32	4	65	55	45
0.032	½	0	3	66	56	46
	⅝	0	3	66	56	46
	¾	0	2	66	56	46
	1	0	3	66	56	46
	1¼	0	3	66	56	46
0.040	½	0	3	67	57	47
	⅝	0	3	67	57	47
	¾	0	2	67	57	47
	1	0	3	67	57	47
	1¼	0	3	67	57	47
0.051	½	0	4	70	60	50
	⅝	0	3	69	59	49
	¾	0	3	69	59	49
	1	0	3	68	58	48
	1¼	0	4	68	58	48
0.064	½	0	4	75	65	55
	⅝	0	4	74	64	54
	¾	0	3	72	62	52
	1	0	3	70	60	50
	1¼	0	4	70	60	50

NOTE: In all joggled or recessed areas, 5 deg must be added to the "Bearer Angle Recommended" for the immediate section on the bearer where these occur. Overlap recommended for joggled sections is the same as for normal sections. If the lower edge of the slanted bearer interferes with complete part forming, or hinders the function of the form block, this edge may be cut out or trimmed.

Table 4 – Specifications for construction of bearer blocks. A bearer block (see diagram at top) is a device consisting of an auxiliary block placed adjacent to the regular form block in the region of the shrink flange

Two other devices are used to a limited extent in this connection, both of which employ the principles of punch press forming dies but are adapted for use with rubber pressure forming. It was found that more economical tooling resulted and the set-up time was no greater than that required for conventional form blocks.

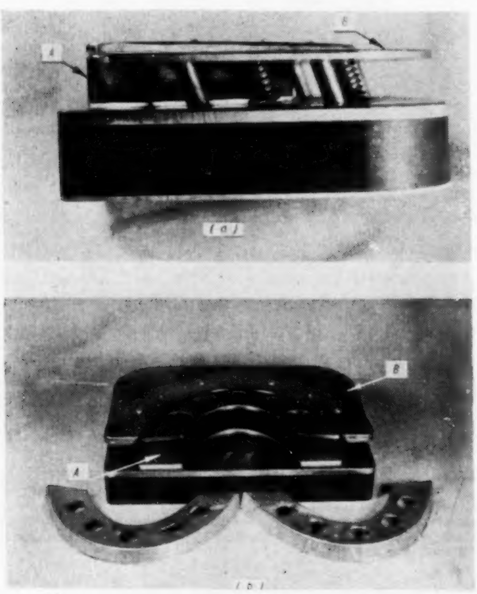
One development along this line consists of a simple push-through die of the type illustrated in Fig. 28. The



■ Fig. 28 – Push-through die for rubber pressure press

die consists of a cavity built up from Masonite (a fiberboard formed at rather high temperature and pressure) and faced with a thin steel plate, as shown in the photograph. The punch is made from Masonite or steel depending on the number of parts required. A part made from 24S-T alclad is shown as it comes from the die. It may be noted that the flanges are free from wrinkles and fully formed. Positive alignment of die, blank, and punch is provided by using locating pins as is shown.

A more elaborate, but none the less useful, set-up is shown in Fig. 29. The die consists of a regular form block



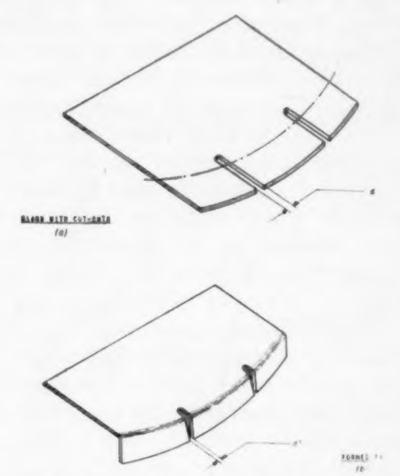
■ Fig. 29 – Modified draw die for rubber pressure press

(A), which serves as a punch, and a pair of movable plates (B), mounted on springs as shown in (a) and fitting closely around the periphery of the form block. The portion of the blank which forms the flange is inserted between the movable plates. The action of the rubber in the press then forces the plates downward against the pressure of the springs, drawing the blank from between the plates and over the side wall of the punch. It may be seen that this permits "drawing" the material as described previously. The springs also serve the purpose of ejecting the formed part when the press pressure is removed.

A part formed on a conventional form block is shown at the right in (b) and a part made on this die may be seen on the left. A trimming operation but no hand forming is required after forming.

Dies of this type cost approximately one-third as much as a standard all-steel double-action die. Such a saving is effected by using Masonite in the punch and base, as well as by decreasing the man-hours necessary to fit the thin steel plates, as contrasted with the material cost and labor required for a standard steel die.

Although these methods are often utilized when flanges having high compressive strains cannot be avoided, methods which obtain the desired contour without actually subjecting the material to such high strains are employed in the majority of cases. This is usually accomplished by using cut-outs in the flange at the proper interval, depending on the sheet thickness, flange width, and radius of contour. The metal flow resulting from this is shown in Fig. 30. A section of a blank with cut-outs is shown in



■ Fig. 30 – Flow of metal in a "shrink" flange with cut-outs

(a) and the part after forming in (b). It may be seen that instead of the metal compressing, it is relieved by the cut-outs, the parallel sides of which move together a distance proportional to the amount of strain which would be required if the cut-out were not there. This is shown by the difference in width between d and d'. A view of a part formed with and without proper cut-outs is shown in Fig. 31.

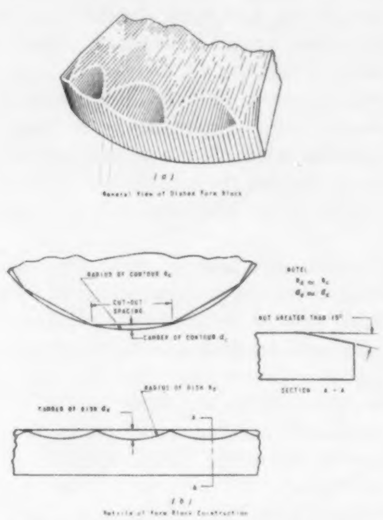
■ Fig. 31 – Effect of cut-outs on wrinkling of "shrink" flanges

Although a great deal of success is possible with this method for "S-O" materials, some compression of the flange is still required unless the cut-out spacing is very small. Thus it was not found practical to form severe concave flanges in 24S-T by this method.

It is obvious that since little or no compression can be obtained in 24S-T without wrinkling, some method must be used which involves bending only. The practical application of this method will be briefly outlined without

reviewing the basic principles, which are described elsewhere.[7]

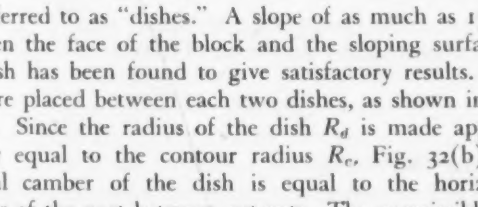
Theoretically, it is possible to form a convex flange without compressive strains by bowing the entire web of the part to the same radius as that of the contour of the flange. However, in practice it has been found possible to accomplish essentially the same results by bowing the web adjacent to the flange, and fairing this into the flat web as shown in Fig. 32(a). The resulting hollow regions



■ Fig. 32 – Dished form block for curved flanges

are referred to as "dishes." A slope of as much as 15 deg between the face of the block and the sloping surface of the dish has been found to give satisfactory results. Cut-outs are placed between each two dishes, as shown in Fig. 32(b). Since the radius of the dish R_d is made approximately equal to the contour radius R_c, Fig. 32(b), the vertical camber of the dish is equal to the horizontal camber of the part between cut-outs. The permissible vertical camber is limited by such design considerations as part interference and allowance for rivet pattern, so this is usually the determining factor in cut-out spacing.

A view of a 24S-T alclad part formed in this manner is shown in Fig. 33. A part formed from 0.025-in. stock on



(a) (b) (c)

■ Fig. 33 – Effect of dished form block on 24S-T parts

a conventional form block, without cut-outs, may be seen in (a) and a corresponding part formed by dishing the block is shown in (b). View (c) shows a corresponding part formed from 0.064-in. stock.

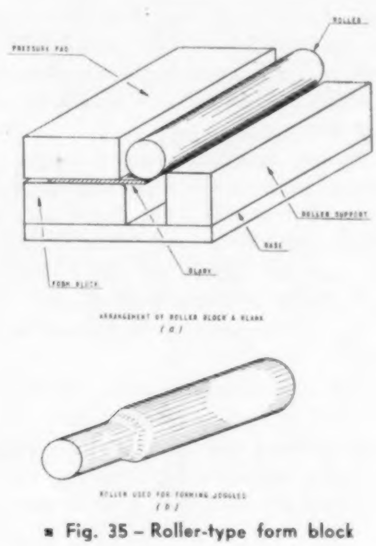
[7] See *Iron Age*, Vol. 149, No. 24, June 11, 1942, pp. 49-53: "Forming Convex Flanges and Joggles," by William Schroeder and R. B. Glassco.

Closely allied to the problem of forming convex flanges is that of forming joggles. A joggle consists of an offset in the bend line of a part, as shown in Fig. 34(a). It may be seen that the forming operation usually involves reducing the width of the web of a part in a local region without changing any other dimensions. Joggles may be formed with mechanical dies, or on the rubber pressure press by using dished form blocks.

A view of the dishing principle applied to joggles is shown in Fig. 34(b). The block is reduced in height at the bend line by an amount equal to the depth of the joggle. Theoretically, this reduction should extend across the entire face of the web, but again it has been found that satisfactory results may be obtained by fairing the reduced section into the web as shown in the figure. The same general considerations which determine the limiting depth of dish permissible for convex flanges apply in this case.

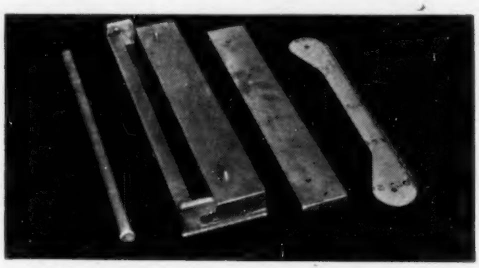
■ Fig. 34 – Joggle in sheet metal part and form block

One additional method of forming both joggles and very narrow straight flanges on the rubber press is to utilize a roller in conjunction with the form block as shown in Fig. 35. The blank is placed as shown in (a) with the roller on top. A pressure pad higher than the roller is



■ Fig. 35 – Roller-type form block

required on top of the blank to ensure the press load striking the web of the part first and holding it in position. The roller is then pressed downward into the groove, forming the flange as it descends. If a joggle is desired in the flange, the roller is shaped with an offset as shown in (b), and the form block undercut to mate with it. Fig. 36 shows a part having a very narrow flange formed in this manner from 24S-T material.



• Fig. 36 – Roller block for rubber pressure press and formed part

A joggle or some other form of indentation is sometimes required in the web of the part and will not form in 24S-T. In such cases, it is often desirable to provide a cover plate over the top of the blank with the desired impression shaped into it.

Summary – (1) This process provides a method of obtaining a reasonably high rate of production of simple flanged parts in an industry which must contend with a very large number of different parts and a relatively small number of each.

(2) Tooling is comparatively inexpensive.

(3) Consistency of results is affected by (a) average pressure on the platen at which the press is operated, (b) proximity of adjacent objects on the table, (c) rubber hardness, and (d) form-block height.

(4) Angular tolerances of ± 2 deg should be permitted for straight flanged parts.

(5) Concave flanges may be formed from 24S-O and 24S-T when the maximum tensile strains do not exceed 10-12%. This may be considerably exceeded in some special cases.

(6) Convex flanges may be formed on simple form blocks from 24S-O, 52S-O, 53S-W, and 61S-W with compressive strains of 1-6% depending upon material thickness and flange width. It is difficult to obtain compressive strains greater than 0.5% in 24S-T.

(7) The effective limits for concave flanges may be increased by using mechanical devices such as push-through dies or a form of draw die. The limits may also be increased by the use of cut-outs and "dished" form blocks.

(8) Joggles can be formed in 24S-T by the use of "dished" form blocks or mechanical rollers.

(9) Very narrow flanges may be formed by using mechanical rollers.

(10) Forming indentations into the web of a part may often be facilitated by using a pressure pad.

(11) Overall forming conditions may be improved and spring-back angles stabilized by adjusting the operating conditions to obtain the maximum possible pressure on the form block.

(12) Pressure may be increased on particular parts by (a) increasing the applied press load, (b) using displacement blocks adjacent to the parts required, and (c) increasing the height of the form block. It should be borne in mind that if an excessive number of displacement blocks or high form blocks are placed on any one loading table the desired results will not be achieved.

■ Use of Double-Action Presses

Probably the most important single type of equipment in use in the sheet metal industry for making deep sections, of which boxes, cups, and dome-shaped parts are examples, is the double-action press. It has long been recognized as the most efficient method of making a large number of any one particular shape, but has been regarded by many as too expensive when a comparatively large number of different parts and relatively few of each are required. The criticisms are made that the set-up time involved in changing dies is quite long and that the individual dies are expensive.

While it is true that both of these objections are valid when mechanical presses are used with steel or iron dies, the set-up time may be reduced to a reasonable figure by the use of hydraulic equipment. Although the production rate after set-up is lower than with the mechanical type, the rate obtainable exceeds any other method of fabricating the above-mentioned parts by such a large factor that this objection may be neglected. This is particularly true when short runs are made.

The second objection, that of high die cost, has been the subject of much research within the past few years. Two types of construction have resulted. The first consists of dies built up of wood, contoured to shape, and faced with a thin steel plate. The second line of development centered around finding some material, other than steel, which would be satisfactory for the small runs (500 to 5000 parts) desired.

Past experience at Lockheed with Kirksite used in drop hammer dies indicated that it had very good wearing qualities. Accordingly, a round-cup draw die was constructed of it, and 77 parts, requiring rather severe forming, were made from 24S-O alclad sheet without any signs of wear, either on the drawing surfaces or draw radius. It was concluded that the die would last for several hundred more parts.

Since that time, over 150 double-acting draw dies have been made from Kirksite for production, with at least one of them used for making over 4000 parts to date.

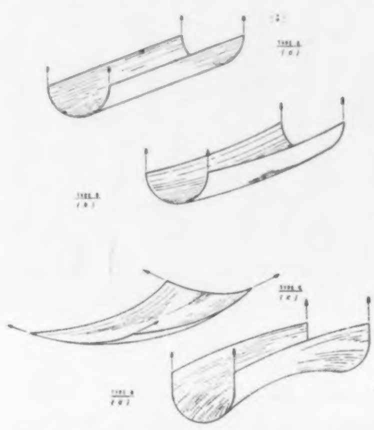
The decreased cost effected by utilizing this material lies in the fact that it is readily reclaimed, foundry technique is simplified due to the low melting point (717 F), and it is easily machined.

These same qualities make Kirksite an ideal material for use as stretch punches. When used for this, the punch is often cored to reduce the total weight. Steel, plaster, or green sand make satisfactory cores.

■ Stretch Forming

Stretch forming on the double-acting press has already been described in the section dealing with general types of forming equipment. The general method of using this process has already been mentioned, but it is felt that a more detailed discussion of the applications and limitations of the process may be of value.

Classification of stretch parts – It may be seen from Fig. 37 that forming by stretching a single-curvature part, such as is illustrated in (a), involves simple bending around the punch plus a uniform tensile strain across the width of the sheet. Parts of this type will be referred to as Type A. On the other hand, when a compound curvature is required as shown in (b), the uniform tensile strain condition is replaced by a differential strain across the sheet caused by the difference of length around the punch from one section to another. It may be seen that one of the factors affecting the forming limits is the difference in strain which the material will withstand between adjacent sections of the blank without failure. This is usually the only limitation imposed on Type B parts exemplified by (b). Parts of this type consist of a rather severe curvature in one direction and are stretched in the direction of the principal curvature as shown.



■ Fig. 37 – Types of stretch-formed skins

On the other hand, when a part consists of relatively shallow double curvature, and the degree of curvature is approximately equal in both directions, or maximum normal to the direction of stretching as shown in (c), the forming is often limited by wrinkling of the sheet. Such parts are grouped in Type C. The cause of wrinkling of this type is under investigation at the present time, but appears to be induced by shear stresses imposed on the material.

Another group of parts which may be limited by wrinkling of the sheet consists of those having maximum strains at the edges of the part and minimum strains at the center. This class is illustrated in (d) and such parts are commonly referred to as "saddle-back" or Type D parts. In this case, the wrinkling is caused by compressive forces induced by the geometry of the part.

The four classes of parts and the factors which determine the forming limits are summarized in Table 5.

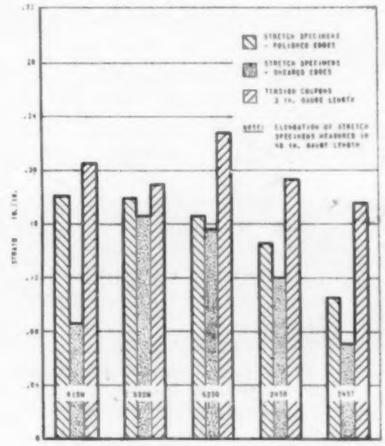
Forming limits for stretching – Although little information has been obtained to date on the limits of wrinkling for Types C and D parts, due to their relatively complex nature, the permissible strains have been investigated for the various materials.

Test specimens were formed over a stretch punch until failure occurred in the sheet and the elongation measured over a 42-in. gage length. Specimens were used with edges "as sheared" as well as edges which were sheared, filed, and then polished with steel wool.

Table 5 – Classification and Limitations of Stretch Parts

DESCRIPTION	TYPE	TYPES OF FAILURE TO WHICH SUSCEPTIBLE	
		FRACTURE	WRINKLING
	A	⊙	
	B	⊙	
	C	⊙	⊙
	D	⊙	⊙

Results of these tests are given in Fig. 38, together with the elongation in a 2-in. gage length as measured from standard tensile coupons cut from the same sheets of material before forming. It may be noted that no correlation exists between the elongation values of the material obtained under the different conditions.



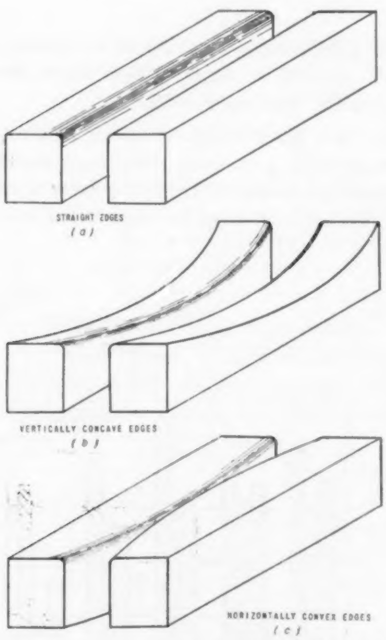
■ Fig. 38 – Elongation of various materials in stretching

The difference in elongation between specimens having sheared and polished edges may be minimized for the majority of cases. This is due to the fact that for parts of Types B or C, the maximum strains commonly occur at the center of the sheet and the minimum at the edges, thereby reducing the effect due to edge condition of the blank. It is only in the case of Type D, or saddle-back parts, that the maximum strains occur at the edges of the sheet. Thus the elongation values given for specimens having polished edges may generally be used for all cases except Type D parts.

It must be realized that the values given represent the *average* of a number of specimens rather than the minimum of the individual specimens. A rather large variation was encountered between the specimens in some cases, particularly for the 24S alloy. Therefore, to allow for material variation as well as to allow for the fact that the strains were measured *after* failure of the sheet, a margin of safety should be employed when using these values.

Techniques in stretch forming on the double-acting press – Experience of the past year at Lockheed, using the double-acting press for stretching, has resulted in the formulation of a number of techniques which have proved useful in applying this process. A number of these will be reviewed briefly.

(1) Supporting rails – For most cases it has been found desirable to use rails having flat clamping surfaces and the inner edges consisting of straight-line elements as shown in Fig. 39(a). These should have a corner radius of at



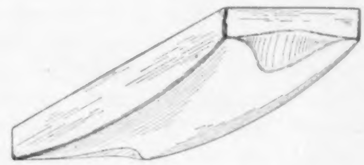
■ Fig. 39 – Types of clamping rails used in stretch forming

least 1 in., over which the material is formed. However, it has been found advantageous in some cases of Class C parts to use rails with concave plates as shown in (b). This curvature should not be greater than the corresponding curvature of the punch, or the maximum strains will occur at the edges of the blank. Plates curved as shown in (c) have also been used with some success. However, the latter practice is not recommended since the same general force pattern is set up which causes wrinkling in saddle-back parts.

(2) Stretch punch – In general it has been found necessary to use a highly polished and well-lubricated surface on the punch to ensure the maximum uniformity of strain distribution over the sheet.

In the case of Type C parts, the desired contour is usually quite flat and the punch must be shaped as shown in Fig. 13. From this, it can be seen that it is necessary to form the material over the corner of the punch. To prevent tearing the sheet at this point, a radius equal to or larger than the rail radius should be used.

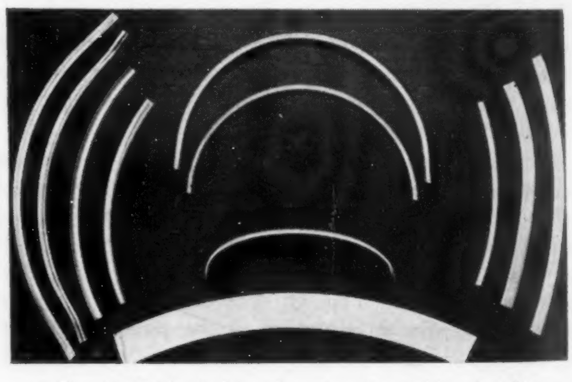
To reduce wrinkling in Type C parts, the ends of the punch may be dished as shown in Fig. 40. This is usually



■ Fig. 40 – Dished punch for Type C parts

done by designing the punch in such a manner that the distance between the rails to which the sheet must conform is nearly equal across the width of the sheet, thereby reducing the shear stress which causes wrinkling.

Special stretching machines – Although a large portion of the work on stretch forming at Lockheed has centered around skin panels and fairings as just described, a considerable amount of work has been done on smaller sections, particularly in the field of curved channels of various shapes. A view of several such sections formed in this manner is shown in Fig. 41.



■ Fig. 41 – Typical channel sections formed by stretching

The principal advantage of forming such parts by stretching lies in the fact that spring-back problems are virtually eliminated, as explained previously. This enables the die maker to shape the form block to the dimensions of the final part rather than having to undercut the block by some experimentally determined quantity to allow for the elastic recovery of the part after the forming load is removed.

Since the forming and clamping forces are comparatively low, even for stainless steel, it is more practical to utilize a smaller piece of equipment which is designed to accommodate the particular type of part in question, instead of the large double-acting press. Such an experimental machine was designed at Lockheed and is shown in Fig. 42. It must be borne in mind that this is not intended as a

production machine, but rather as a means of developing and proving tooling for the type of parts shown in Fig. 41. Since the clamping loads are quite small, mechanically actuated jaws are used.

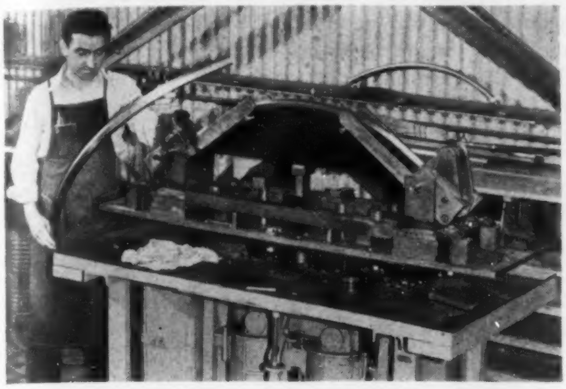


Fig. 42 – Experimental stretching machine for making curved channel sections

The specimen before stretching may consist of a flat strip or a particular shape preformed on the power brake. The forming strains required usually determine which is used. In the latter case, special jaws must be made for holding the part. In general, either Masonite or Kirksite punches are used.

Summary – (1) Stretch forming is limited to parts which do not have severe double curvature.

(2) Forming limits of skins and fairings are determined by the tensile strains which the material will withstand or by wrinkling of the sheet. Both are determined by the geometry of the part.

(3) Considerable difference in maximum obtainable elongation may be obtained for materials with different edge conditions when the maximum strain occurs at the edge of the sheet. However, edge condition has little effect on the majority of parts since the maximum strains usually occur in the central portion of the sheet.

(4) Elongation in a 2-in. gage length obtained from standard tensile coupons are not indicative of the average elongation obtainable in stretch forming.

(5) It is desirable to use a double-acting press for stretch forming large skins and fairings because of the high clamping forces available. This permits the use of flat clamping plates rather than serrated jaws. The latter tend to cause stress concentrations within the clamps, which often result in tearing the sheet during forming.

(6) In general, it has been found that flat rails consisting of straight-line elements at the inner edges are the most practical. However, it is sometimes advantageous to use curved rails.

(7) The radius on the edge of the rails and punch should be at least 1 in., and larger if possible.

(8) The punch should be highly polished and lubricated to equalize the strain distribution along the length of the sheet.

(9) Curved channel sections may be readily formed by stretching, thereby eliminating tooling development problems due to spring-back in materials of harder tempers. For these parts, a small stretching machine is usually sufficient.

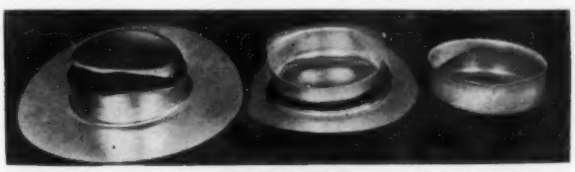
Drawing on a Double-Acting Press

The mechanics of drawing has already been mentioned briefly. It will be recalled that this term applies to a combined strain condition consisting of elongation in a radial direction and compression circumferentially. The only geometrical shape in which this condition occurs ideally is that of a cylindrical cup. It may be seen that, with this shape, the stress and strain condition is the same at any point a given distance from the center of the bottom of the part.

Since the primary stresses and strains occurring in drawing are compressive, and the tensile elongations a secondary phenomenon which results from this compression, a cup is the most severe shape for this type of forming. On other geometrical shapes, such as boxes, the compression is relieved to a certain extent by the metal flowing from the corners (which may be regarded as segments of a cup) into the flat sides.

It has been pointed out that the ratio of cup radius r to blank radius R may be taken as an index of the severity of the forming operation involved. However, measurements of a large number of parts indicate that the area of a formed cup is the same as the area of the unformed blank within an accuracy of about 1%. This will be referred to as the "principle of constant area" for the remainder of this report. By utilizing this fact, it then becomes clear that the ratio of cup height h to cup radius r is also an index of the severity of the forming operation. Since the latter is a more useful quantity for both designers and shop men, it will be used throughout this paper. The largest value of h/r to which any given material may be formed without failure is called the "limiting" h/r. Limiting values may be used as an index of the formability of the various materials.

During the course of a series of tests at Lockheed to determine the limiting values of h/r for various aluminum alloys when formed in one stage, three different types of failure were encountered. Type I (Fig. 43a) is the most common, in which the bottom of the cup is broken out as shown. This failure occurs due to the tensile stresses in the wall of the cup exceeding the ultimate strength of the material.



(a) Type I (b) Type II (c) Type III

Fig. 43 – Types of failures occurring in drawn parts

Failures of this type occur in 24S-O, 52S-O, and 53S-W.

Failures such as that shown in (b) are classified as Type II. They have been found in 24S-T material when blank sizes *considerably larger* than the limiting size are formed. Fractures of a similar nature have also been observed in redrawn parts and boxes. These are characterized by the break starting at the lower tangent point of the draw radius as shown in (b) and appear to be a function of the rate of work-hardening as well as the ductility of the material.

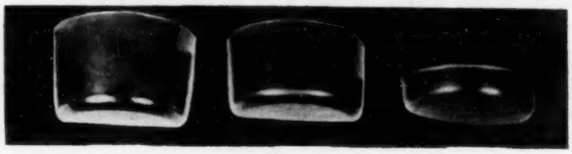
Type III failures occur as illustrated by the 24S-T cup shown in (c). In this case, the side wall of the cup fails

longitudinally starting at the upper edge. The cause of this phenomenon is under investigation at the present time. It is believed to be associated with failure of the material in shear along diagonal planes.

In addition to failures of this nature in 24S-T, the same type of failure has been observed in cups drawn from 52S-½H, and 61S-W when blanks *slightly* larger than the limiting size are formed.

Maximum depths of 6-in. diameter cups which can be formed from the various aluminum alloys are shown in Fig. 44, together with the limiting values of h/r for each.

52S-0 h/r = 1.50 | 24S-0 h/r = 1.50

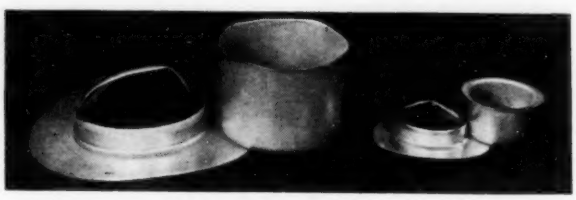


53S-W h/r = 1.43 | 52S-½H h/r = 1.28 | 24S-T h/r = 0.44

■ **Fig. 44 – Limiting values of cups drawn in one operation**

These h/r values have been computed on the basis of the blank size used, assuming square corners on the bottom of the part. The actual average depths obtainable will be slightly larger due to the radius on the bottom of the punch.

Additional cups were formed on a 3-in. cup die to determine whether the limits previously determined were valid for different sized cups. The limiting sizes for 24S-O parts determined on both dies are shown in Fig. 45. It



h/r = 1.6 h/r = 1.5 2r = 6 in. t = 0.051 in. | h/r = 1.6 h/r = 1.5 2r = 3 in. t = 0.025 in.

■ **Fig. 45 – Effect of reduction of size on the limits of drawn cups**

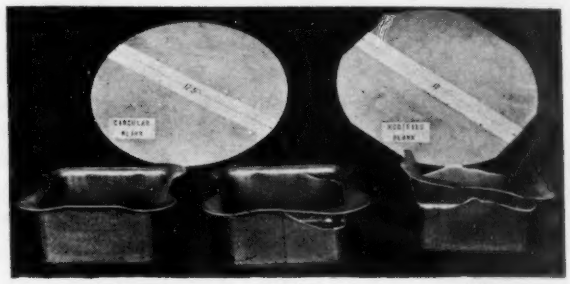
may be noted that good parts were formed on both at $h/r = 1.5$ and parts failed on both at $h/r = 1.6$. Blank thicknesses of 0.020 in. and 0.040 in. were formed on the small die to the same depth. From these results, it may be concluded that within the range tested there is little or no scale effect present with these alloys.

Objections have been made that tests on drawn cups are not representative of the general shapes usually required in production. However, it is felt that such tests do provide a method of comparing the formability of the various alloys. Since the cup represents the most severe condition, the results are useful as an index of the conservative limits to which parts can be formed; that is, all other shapes can be formed at least as far, and in many cases much farther. Such results may be applied to more complex shapes by the proper breakdown of the parts into segments of a cup.

Drawn boxes – However, since it was realized that forming conditions were changed considerably by changing the shape of the part, a program was initiated to investigate the limits for drawn square boxes having rounded corners. This shape was selected since it is one of the simplest shapes in which the strain pattern is not uniform around the perimeter of the part.

Work done to date indicates that the limits determined for cups are very conservative when applied to boxes. Furthermore, the limiting depth to which boxes may be drawn is not a function of the material only, but also of such factors as (1) corner radius of the box, (2) width of flat sides between corners, (3) blank shape, and (4) grain direction of the blank.

The two parts shown in (a) and (b) of Fig. 46 illustrate



(a) (b) (c)

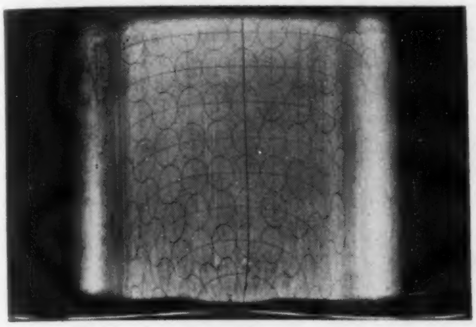
■ **Fig. 46 – Effect of grain direction and blank shape on limits of drawn boxes – (upper left) circular blank, 12.5-in. diameter; (upper right) modified blank, 12 in. across straight section; (a) grain direction parallel to side – circular blank; (b) grain direction 45 deg to side – circular blank; (c) grain direction paralllel to side – modified blank**

the importance of grain direction on the limiting depth of draw. Blanks for both parts were cut from the same sheet of material and formed under similar conditions except that the direction of rolling the sheet (grain direction) was placed in different positions with respect to the sides of the box. In (a) the grain direction was parallel to the side of the box, while in (b) the grain direction made an angle of 45 deg with the sides of the box.

The role played by proper blank development is shown by a comparison of (a) and (c), Fig. 46. The former was formed from a circular blank 12.5 in. in diameter, while (c) was drawn from the modified blank shown above it. The distance across the diagonal of the blank, which formed the corners of the box, was ½-in. smaller with this modification than the circular blank. However, a successful part was formed from the circle, while the modified shape failed as shown even though it was smaller. A large number of other blank shapes have been tried with similar results. Thus it may be seen that shape as well as size of the blank is important.

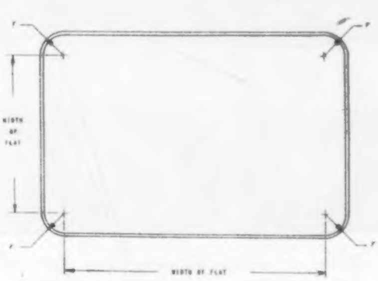
A clue to the reason for this may be found from a study of Fig. 47. A grid pattern consisting of squares and circles was photo-printed on the blank before forming.[1] The distortion due to forming is clearly revealed in the photograph. It may be seen that the lines which were originally parallel are moved together at the top of the part due to the compression of the metal in the corners forcing the flat sides to compress. The "ears," resulting at the corners when circular blanks are used, apparently aid

the general forming by causing this effect to be increased, thus increasing the depth of part which may be formed by decreasing the strains in the corners. This is borne out by the fact that the strains in the corners reach a maximum value for this box at a depth of about 3 in.; that is, increasing the depth of the part to 6 in. by using a circular blank does not result in a corresponding increase in strains.



■ Fig. 47 – Strain relief caused by the flat sides of a box

The distance between the centers of radii of the corners of a square box will be referred to as the "width of flat," L, in the side of the part. This is shown in Fig. 48. The



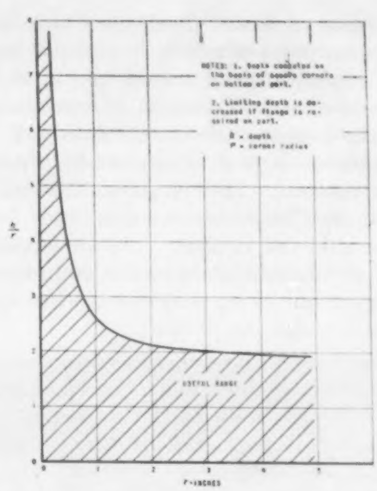
■ Fig. 48 – "Width of flat" in drawn boxes

precise effect on the limiting depth to which a part may be formed by varying the width of the flat has not been determined to date. However, the foregoing discussion indicates that the limiting depth may be varied considerably by changing this distance. It is probable that the depth may be increased by increasing the width up to some critical value of L, beyond which no further increase in depth may be accomplished.

A series of tests on 24S-O material indicates that the severity of draw may be varied appreciably by changing the corner radius. Such tests were made using relatively large widths of flat to eliminate this variable as much as was practicable.

Since it is felt that the formability index, h/r, used for cups is the most practical for general use, this quantity will be used in a similar manner for boxes. It must be realized, however, that it is no longer a direct function of the strains in the material when used for such shapes.

The limits of forming boxes from 24S-O alclad are given in Fig. 49. This curve was constructed from data obtained



■ Fig. 49 – Limits for drawn boxes –24S-0 alclad material

on production dies with corner radii varying from ¼ to 4 in., the majority of tests being made on dies having corner radii of ¼ to 1 in. Designating the corner radius by r, all parts have a flat area of $L \geq$ or $2r$. The limiting values shown presuppose the use of properly designed and constructed dies as well as intelligently developed blanks.

During a different series of tests made on 6-in. square boxes, parts were formed of approximately equal limiting depths for both ⅝- and 1½-in. corner radii, as shown in Fig. 50. Thus it appears that the limiting h/r value for boxes varies with corner radius for both materials tested, namely 52S-O and 24S-O. However, since the overall dimensions of the boxes were held constant and the corner radius varied, the width of flat also changed. The decrease in limiting h/r found for the 1½-in. radius boxes as compared with the ⅝-in. part may be partly due to the decrease of the width of flat as well as to the increase of corner radius.

It is significant that 24S-O and 52S-O have about the same limiting h/r values for cylindrical cups drawn in one operation, but widely differing limits in the case of boxes, as shown in Fig. 50.

24S-0	24S-0	52S-0	52S-0
$r = 1\frac{1}{2}$ in.	$r = \frac{5}{8}$ in.	$r = 1\frac{1}{2}$ in.	$r = \frac{5}{8}$ in.

■ Fig. 50 – Limits of 6 x 6-in. boxes drawn from circular blanks

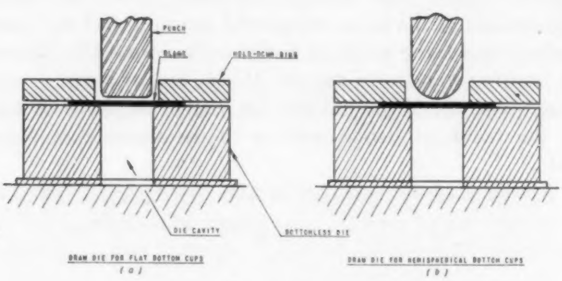
The principle of constant area described for drawn cups applies with approximately the same accuracy when used for boxes. The primary application of this principle for such shapes lies in the field of preliminary blank development. When used in this connection, the part to be formed is divided into sections and the blank area and shape computed for each. These segments may then be laid out in their proper relationship to each other. Where abrupt discontinuities occur between adjacent segments, they should be faired into a smooth curve.

This method of blank development has proved quite useful in some cases where it is essential to reduce the press time required for die try-out to a minimum. However, the user must be cautioned against using re-entrant curves in the outline of the blank since such a condition usually constitutes a more serious handicap than an excess amount of material. This is particularly true of severe corners with their center of curvature inside the outline of the part, as in the case of boxes. The use of excess material in certain places to control the strains in the corners, as has been described for boxes, may be used to advantage by persons familiar with the process.

It must be borne in mind that this procedure is intended only as a method of determining the blank shape on the first strike and is not set forth as a method of determining the final shape. This can only be determined by try-out on the die for which it has been designed. The final shape is dependent in part upon such factors as (1) adjustment of the die itself, including clearances and alignment, (2) lubricant used, and (3) economic considerations including material conservation and method of cutting out the blank.

Drawing hemispherical parts – The third type of parts encountered in drawing is that which is partly unsupported by the punch during the majority of the drawing operations. Parts of this type are formed by a combined drawing and stretching, rather than simple drawing, as in the case of cups. The most simple example of this case is a part having a hemispherical or dome-shaped bottom.

It may be realized from Fig. 51(a) that in drawn parts



■ **Fig. 51 – Effect of punch shape on puckering**

having flat bottoms, the material in the bottom remains practically undeformed. In general, the strains which do occur are tensile and do not tend to cause buckling. Actually, there is a small region between the punch and draw radius which is subjected to combined tension and compression, but due to the support given the blank by the punch, buckling of the blank does not occur.

On the other hand, if the flat-bottomed punch is replaced by one having a hemispherical-shaped end, such as is shown in (b), the portion of the blank inside the draw radius will gain partial support against buckling by contact with the punch at the center only. Thus, the free area is permitted to buckle during drawing. Although this phenomenon, as well as wrinkling of the blank under the hold-down pad, is due to circumferential shrinking of the blank required for drawing, they should be kept distinct, since a different technique is required to correct each.

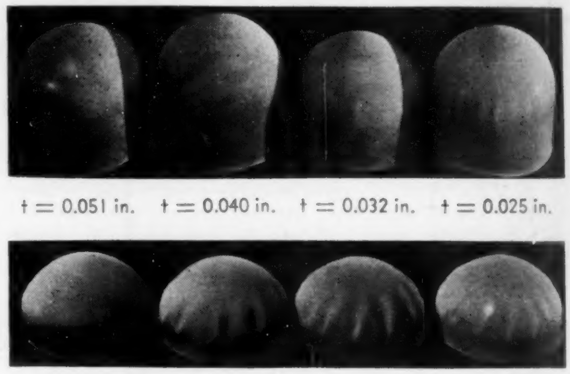
Buckling of the type just described will be referred to as "puckering" and applies to that portion of the blank which is inside the draw radius, while "wrinkling" refers to buckling of the blank while it is still in position between the die and hold-down pad. Wrinkles may be pulled into the die after forming under the hold-down pad but these do not constitute puckers.

It may thus be seen that parts of this type are limited by both the maximum strains the material will withstand, as well as the tendency to pucker.

Puckering may be prevented or reduced by two methods: (1) by increasing the blank thickness, thus increasing the stiffness of the material, or (2) by subjecting the blank to increased radial tension, thereby opposing the tendency to buckle.

The latter must be accomplished by increasing the load on the punch required to form the part. It has been pointed out that this load is a function of the size blank which is being drawn and of the friction forces exerted on the blank by the die and hold-down pad.

A series of tests has been made on such parts having a 3-in. spherical radius, using a bottomless die. Typical parts drawn from 24S-O alclad are shown in Fig. 52. The parts



■ **Fig. 52 – Effect of blank diameter and material thickness on puckering of hemispherical parts**

shown at the bottom of the photograph were drawn from 9-in. diameter blanks using a hold-down load of 36 tons. It may be seen that puckering decreased with increased blank thickness, but even parts formed from 0.051-in. blanks have an uneven surface. However, when the size of the blanks was increased to a value just below that causing failure, slight puckers occurred in parts formed from 0.025- and 0.032-in. material but were completely eliminated in 0.040- and 0.051-in. thick parts.

Although it is possible that this defect could be decreased in the small blanks if sufficiently high hold-down pressures were used, it appears that puckering can be more effectively controlled by increasing the blank size in parts where this will increase the tensile stresses in the part.

To determine the limits of puckering of similar parts of a different size, a die was constructed having a cup diameter of 10 in. and a 5-in. spherical radius on the punch. Good parts were formed on this die from blanks 0.064-in. thick. Thus it appears that the limits of puckering are a function of the blank thickness t and the spherical radius R'. Expressing this as a ratio, it is seen that under the most favorable conditions (large blank sizes) the critical value of R'/t is 75 for the smaller part and 78 for the 5-in.

spherical radius. This small difference is not significant since the blank thicknesses were necessarily varied in rather large steps.

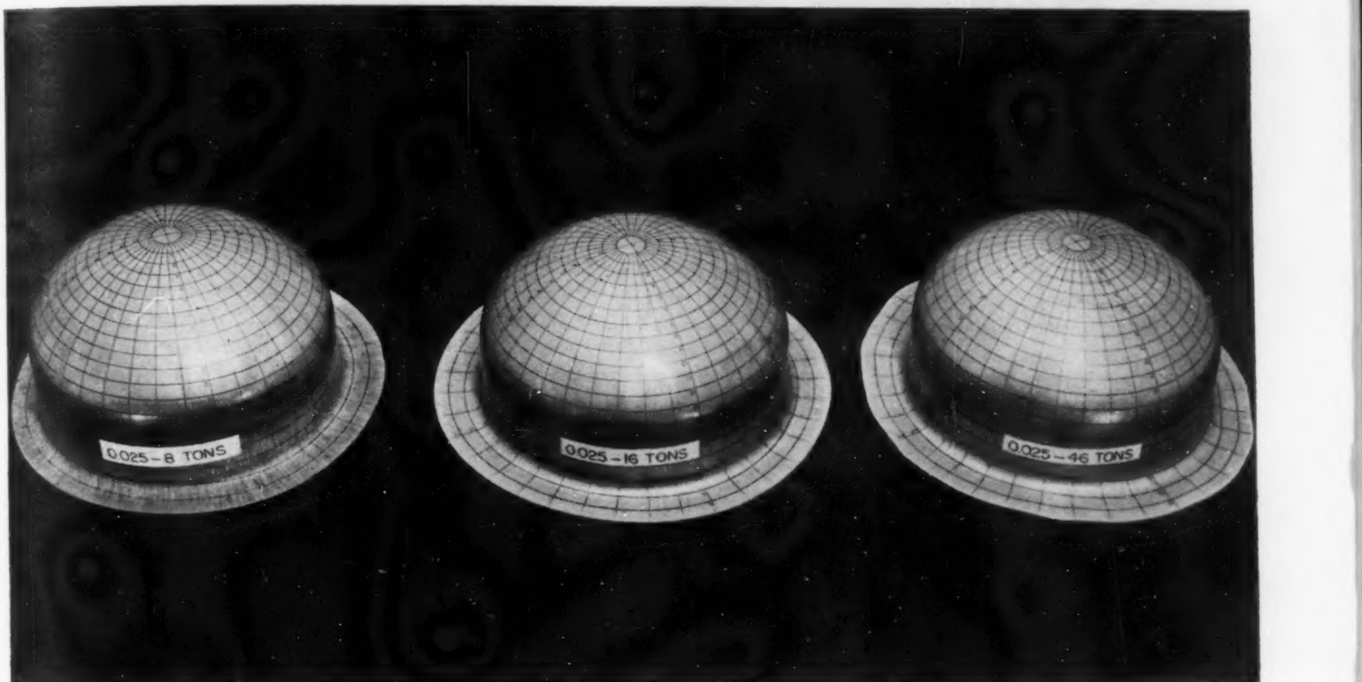
These results indicate that any part having an R'/t ratio of 75 or less may be formed on a bottomless die without puckers when the proper technique is used. On the other hand, small wrinkles may be pushed out on a well-mated bottoming die at values considerably higher. Under such conditions, $R'/t = 120$ may be used.

Parts formed from 52S-O blanks appear to have the same puckering characteristics and limits as 24S-O.

a series of dies and reduce the blank in stages until the desired shape is obtained. Little quantitative information is available on this subject applicable to the aluminum alloys commonly used in aircraft construction.

In order to obtain some indication as to the ultimate depth to which the various alloys could be drawn, it was desired to make multiple-stage cupping tests, thereby establishing some quantitative values which may be used in design calculations.

Through the courtesy of the Norris Stamping Co. in Los Angeles, a seven-stage die set-up was made available.



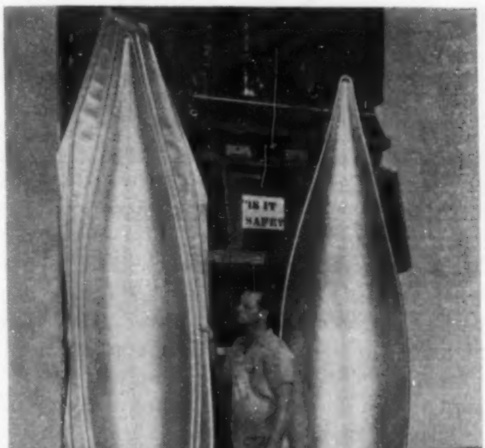
■ Fig. 53 – Increased hold-down load (right) eliminates puckering, as seen in left and center parts

When the spherical radius on the punch is increased to 3½ in. and the punch diameter remains the same, thus flattening the dome to only a portion of a hemisphere, the tendency to pucker is considerably reduced. Fig. 53 shows three parts formed over such a punch. It may be seen that the parts formed with 8- and 16-tons hold-down present an uneven surface in 0.025-in. blanks. However, when the hold-down load is increased to 46 tons, a perfect part results. This is an R'/t value of 120.

Thus it is indicated that the critical R'/t ratio is influenced by the r/R' ratio in which r is the cup radius and R' indicates the spherical radius as before.

Although the number of actual parts to which these results may be applied directly is rather limited, the general principle involved may be used in many practical applications. One such case is that shown in Fig. 54 in which the same type of drawing on an unsupported blank is involved. In this case, tension was maintained in the critical sections by the use of beads which exert a drag on the material flowing into the die. However, the general principle involved is identical. It is of interest to note that a bottomless die was used in forming the part shown in Fig. 54.

Redrawing – Since many parts are too severe to be formed in one operation, it is sometimes necessary to make



■ Fig. 54 – Example of unsupported drawing using beads to increase radial tension on the blank

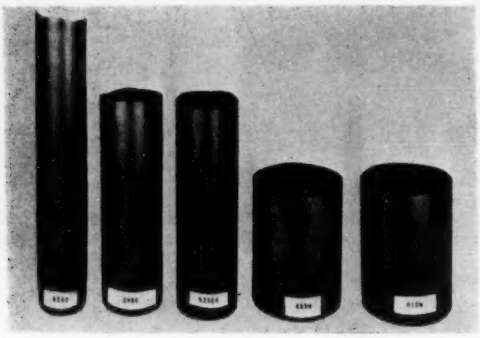
These dies were designed for drawing 1.75-in. diameter tubes from 9.2-in. diameter blanks of 3S-O aluminum. A view of the parts as they come from each stage is shown in Fig. 55. The diameters and h/r values are shown below each.

(a) (b) (c) (d) (e) (f) (g) (h)

(a) $2r = 1.75$ in. $h/r = 13.3$ (b) $2r = 2.00$ in. $h/r = 9.80$ (c) $2r = 2.35$ in. $h/r = 6.90$
(d) $2r = 2.75$ in. $h/r = 5.02$ (e) $2r = 3.40$ in. $h/r = 3.15$ (f) $2r = 4.25$ in. $h/r = 1.86$
(g) $2r = 5.70$ in. $h/r = 0.80$ (h) $2r = 9.20$ in. Blank

■ **Fig. 55 – Stages used in multiple-cup tests**

The apparent limits for the various materials tested are given in Fig. 56 together with the maximum size cup which could be formed without obvious failure of the part. It may be noted that 52S-O gave the best results while 24S-O and 52S-½H were next. Less depth could be obtained with 53S-W and 61S-W than with the others.



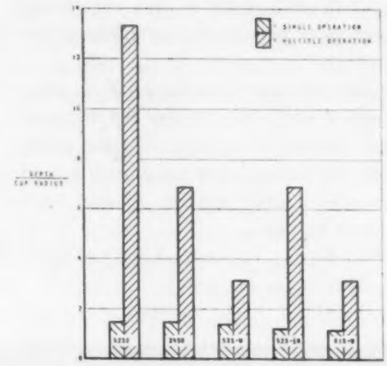
52S-0 $h/r = 13.3$ 24S-0 $h/r = 6.90$ 52S-½H $h/r = 6.90$ 53S-W $h/r = 3.15$ 61S-W $h/r = 3.15$

■ **Fig. 56 – Limits of redrawing various aluminum alloys**

In comparing the depth of draw obtained from the various alloys, it is necessary to remember that the reductions per stage were selected on an arbitrary basis, guided by practical experience. They do not necessarily represent the best combination of reductions.

As has been mentioned, the parts shown do not have any obvious failures. However, a severely worked surface condition is present on some and has not been taken into consideration. The effect of this condition on the strength of the part, as well as methods of eliminating it, is under investigation at the present time. Therefore, the limits as shown should be used with caution.

However, it is felt that these values do give some indication of the relative formability of the various materials under such conditions. The limits of forming in one operation and redrawing are plotted in Fig. 57 for comparison, but it must be remembered that the peculiar surface defects are not considered in this comparison.



■ **Fig. 57 – Comparative limits for single- and multiple-operation cups**

Summary – (1) Drawing is a forming operation involving a combined stress and strain condition consisting of elongation of the sheet in a radial direction from the part and compression circumferentially.

(2) The area of the formed part is essentially equal to that of the unformed blank from which it was produced.

(3) The ratio h/r is used as an index of the severity of the forming operation performed where h = depth of the formed part and r = radius of the part after forming.

(4) Three types of failures have been encountered during forming tests: (a) the bottom of the part is pushed out by the punch, (b) the part fails at a point tangent to the draw radius, and (c) the side wall of the part splits.

(5) Within reasonable limits, no scale effect could be detected for 24S-O alclad material. Cups of 6-in. diameter formed to the same limiting values of h/r as 3-in. diameter cups.

(6) Cylindrical cupping tests serve as a conservative index of the drawing limits for materials, but do not necessarily indicate the maximum h/r values to which they may be deformed in shapes which involve drawing of only a *portion* of the total area.

(7) Limits for drawn boxes are influenced by such factors as corner radii, width of flat sides, blank shape, and grain direction of the blank.

(8) Parts which are given little support against puckering due to the contour of the punch are limited by the tendency to buckle as well as by the magnitude of the strains required. Parts having hemispherical or dome-shaped bottoms are examples of this type.

(a) In such parts, puckering may be controlled by varying blank thickness or by increasing the radial tension on the blank. The latter may be accomplished by increasing the blank size or increasing the friction forces acting on that portion of the blank which is in position between the die and pressure pad.

(b) The limits of puckering expressed as R'/t, where R' = spherical radius on the punch and t = blank thickness, are the same for $R' = 3$ in. and $R' = 5$ in.

(c) Puckering tendency is decreased by increasing the spherical radius without changing the cup radius.

■ Acknowledgment

In the experimental results which are presented in this paper, practically all the members of the Production Research Group employed in sheet metal forming research have made some contribution. However, particular acknowledgment is due to Mrs. M. M. Rockwell for her assistance in preparing the paper, and to Leon Dame and William Box for their assistance with experimental work.

The ELLIOTT-LYSHOLM SUPERCHARGER

by ALF LYSHOLM[1], RONALD B. SMITH[2], and W. A. WILSON[3]

TODAY, the application of supercharging to all types of engines, and particularly to diesel engines, is becoming more and more widespread. Of the many schemes for supercharging, the mechanically driven positive-displacement compressor is very effective for the small high-speed engine and for the diesel engine that requires high torque at low speed.

The authors illustrate the advantages of the Elliott-Lysholm supercharger by describing a 400- to 500-cfm unit.

■ ■ ■

THE AUTHORS: ALF LYSHOLM is chief engineer of Aktiebolget Ljungstrom Augturbin, Sweden. A graduate of the Royal Technical University, he has done fundamental work in the development of the Ljungstrom double-rotation steam turbine, heat exchangers, compressors, and hydraulic torque converters. Mr. Lysholm holds more than 100 patents, many of fundamental technical importance. RONALD B. SMITH, a graduate of the University of Washington, is now director of the Engineering Research and Development Department of the Elliott Co. He has also carried on graduate study at several of the technical schools of this country. His activities have been directed to the design and development of steam turbines and electrical machinery. Before joining Elliott Mr. Smith was with the Westinghouse Electric & Mfg. Co. W. A. WILSON, mechanical division engineer of the Engineering Research and Development Department of the Elliott Co., like Mr. Smith, was formerly with the Westinghouse Electric & Mfg. Co. He is a graduate of Columbia University.

OUR national emergency has served to accelerate the application of supercharging to internal-combustion engines. While the extreme in the utilization of auxiliary charging is, by virtue of necessity, associated with the gasoline aircraft engine, the more moderate application of the scheme has resulted in higher specific output for all types of engines. To the diesel-engine designer, supercharging is particularly significant, since it has not yet been possible to achieve as effective use of the air charge in compression-ignition engines as in spark-ignition types.

Of the several schemes for supercharging now in practice, the mechanically driven positive-displacement compressor has much to recommend it, particularly on small high-speed engines, where simplicity and cost are paramount. For diesel-engine applications where high torque is a necessity at reduced speed, the displacement compressor is effective inasmuch as the engine behaves as an air absorber with about the same speed delivery characteristics as the compressor, with the result that a full air charge can very nearly be maintained at all operating speeds.

There are a number of incentives for the development of a more efficient supercharging compressor. Until now the centrifugal machine, with an adiabatic efficiency of from 70 to 75%, has held a margin in this respect. While the efficiency of the mechanically driven supercharger bears a rather obvious relation to the friction loss of the engine, it has an equally important influence on the charge delivered to the cylinder. An engine with a displacement of D and a compression ratio of ϵ, supercharged to a pressure ratio of ρ and unscavenged, will absorb per stroke an air charge of

$$\frac{D\eta_v}{v_1}\rho\left[1+\frac{1}{\epsilon-1}\left(1-\frac{1}{\rho}\right)\right]\left[\frac{1}{1+\frac{\Delta T}{T_1}}\right]\text{lb}$$

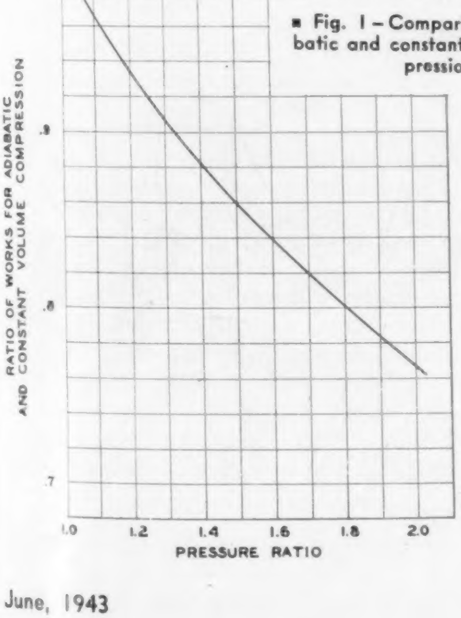
where η_v is the volumetric efficiency of the normally aspirated engine, v_1 and T_1 are the specific volume and the absolute temperature of the atmospheric air, and ΔT is the temperature rise in the compressor. In the derivation of this relation, it is assumed that the heat transferred to the inducted charge by the manifold and the cylinder walls results in a constant-temperature increase irrespective of the charging conditions. The temperature rise in the compressor proper is inversely proportional to its efficiency, with the result that a smaller charge is inducted and a higher compression temperature is attained in an engine operating with an inefficient compressor. For a 4-cycle engine charged to 6 to 7 psi, an increase in the adiabatic blower efficiency from 60 to 70% increases the output by 1.2% through a direct reduction in power absorption, and increases the inducted charge by 2.2%. With the engine maintaining a constant ratio of excess air, the combined yield is slightly over 3% in excess power. Under part-load operating conditions, the effect is an even larger percentage. A more efficient compressor becomes increasingly advantageous when cylinder scavenging is introduced inasmuch

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

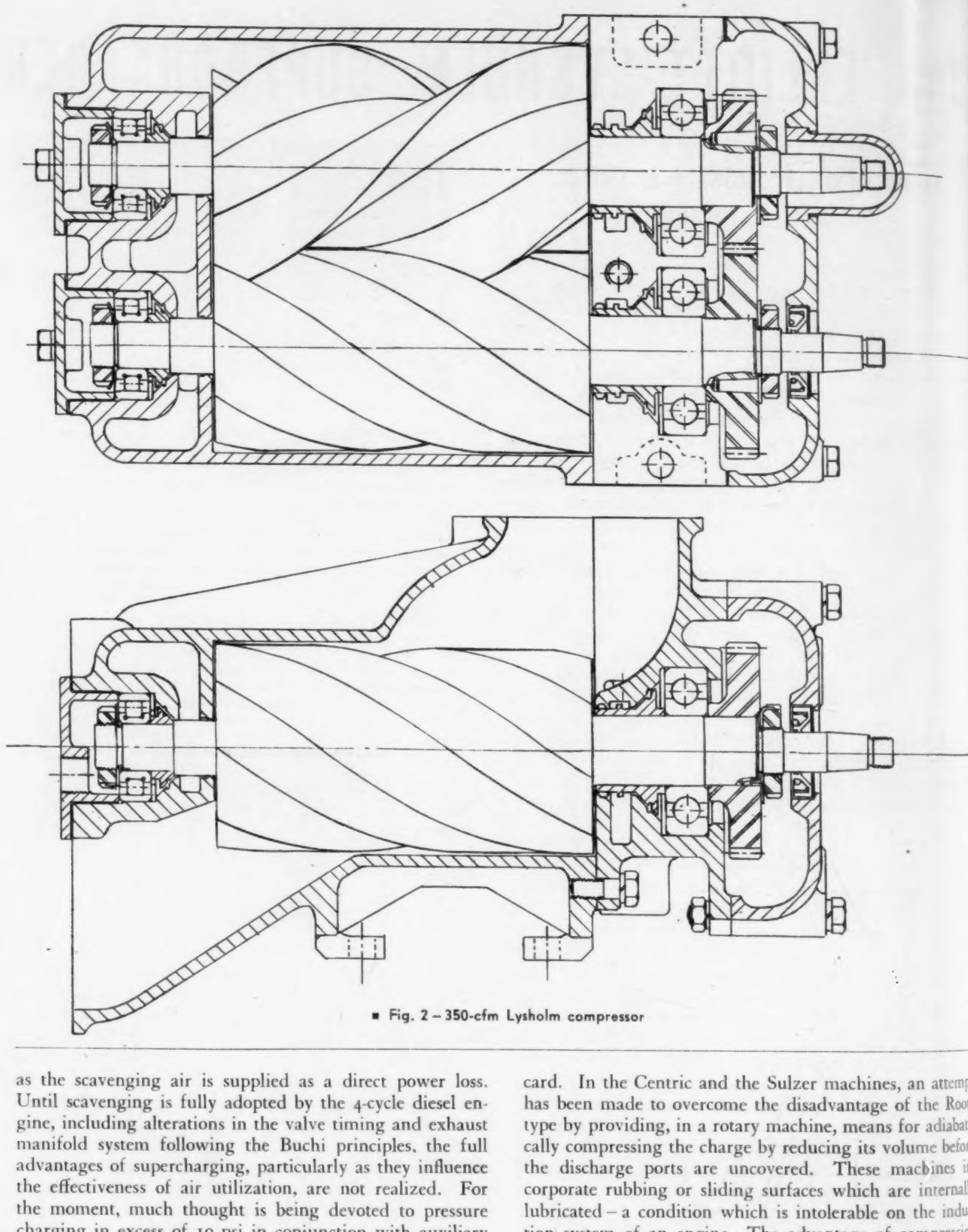
[1] Chief engineer, Aktiebologet Ljungstrom, Augturbin, Sweden.

[2] Manager of research engineering, Elliott Co., Jeannette, Pa.

[3] Mechanical division engineer, Development and Research Department, Elliott Co., Jeannette, Pa.



■ Fig. 1 – Comparison of adiabatic and constant-volume compression



■ Fig. 2 – 350-cfm Lysholm compressor

as the scavenging air is supplied as a direct power loss. Until scavenging is fully adopted by the 4-cycle diesel engine, including alterations in the valve timing and exhaust manifold system following the Buchi principles, the full advantages of supercharging, particularly as they influence the effectiveness of air utilization, are not realized. For the moment, much thought is being devoted to pressure charging in excess of 10 psi in conjunction with auxiliary starting means, and under these conditions the advantages of high compressor efficiency are even more obvious.

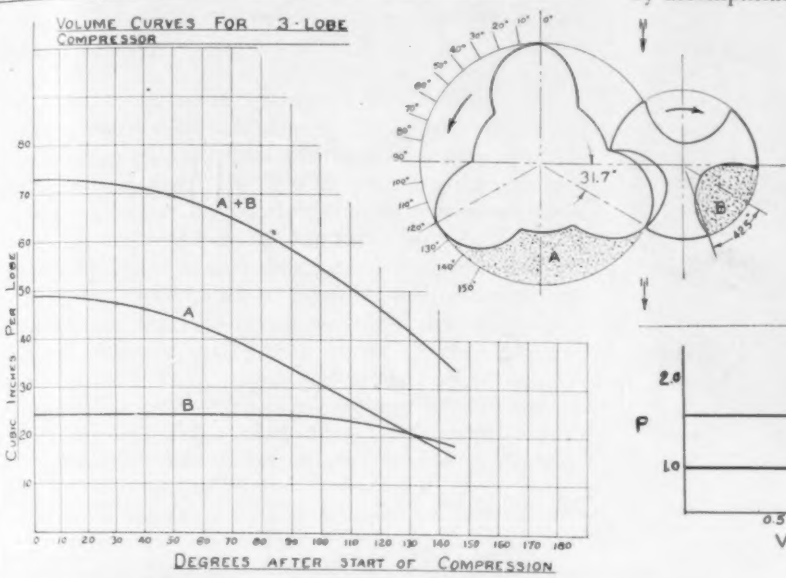
At present, the most common form of positive-displacement supercharger is the Roots-type blower, in which the air is trapped between the lobes and carried around to the exit where a back rush of air from the discharge chamber raises the pressure of the charge. The result is essentially a constant-volume pressure rise and a square-type indicator card. In the Centric and the Sulzer machines, an attempt has been made to overcome the disadvantage of the Roots type by providing, in a rotary machine, means for adiabatically compressing the charge by reducing its volume before the discharge ports are uncovered. These machines incorporate rubbing or sliding surfaces which are internally lubricated – a condition which is intolerable on the induction system of an engine. The advantage of compression within the cylinder is evident from the work ratio for an ideal adiabatic compression to that for constant-volume compression

$$\frac{W_{AD}}{W_{v\,=\,\mathrm{Const.}}} = \left(\frac{k}{k-1}\right)\left(\frac{\rho^{\frac{k-1}{k}} - 1}{\rho - 1}\right)$$

where k is the ratio of specific heats and ρ the pressure

ratio of compression. This function is a comparative measure of the internal efficiency of the two operating cycles and is plotted in Fig. 1. With the 7 to 8 psi charging pressure commonly employed, there is a 14% inherent advantage to the adiabatic cycle, and with the 15 to 20 psi pressures now being contemplated, a 25% gain is a potentiality.

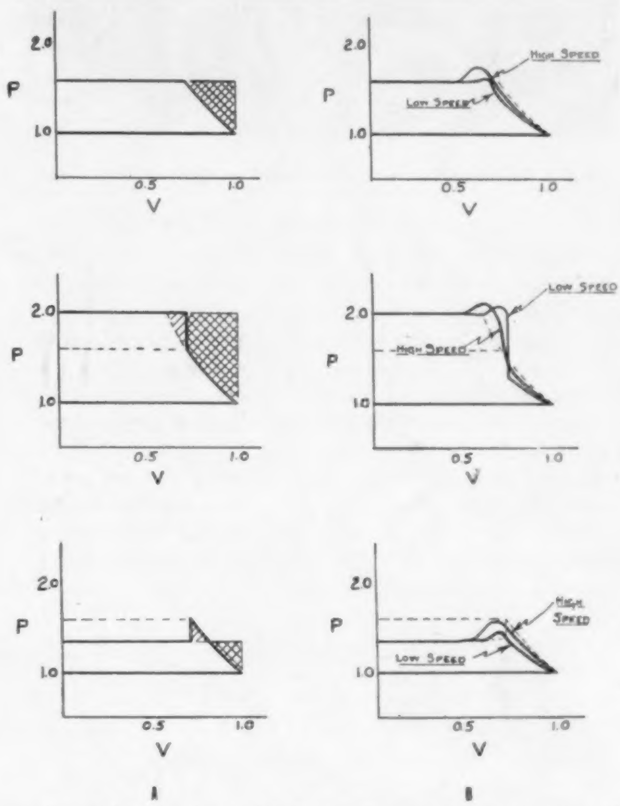
The Elliott-Lysholm rotary compressor was developed in an attempt to overcome the obvious shortcomings of vane- and reciprocating-type compressors by combining the advantages of high rotating speed, simplicity, freedom from rubbing contact, and internal lubrication, inherent in the Roots machine, with the desirable characteristic achieved by internal compression. The development was undertaken in Sweden in 1934 and is covered by a number of foreign and domestic patents.[4] Approximately 30 machines have now been built and tested, varying in size from 15 to 10,000 cfm.



■ Fig. 3 – Compression relationships for Lysholm rotors

A typical example, embodying the Lysholm design features, is shown in Fig. 2. In this compressor, the air is transported diagonally by virtue of the pair of helically lobed rotors which interact to provide an axial as well as a cross seal. The distinguishing feature of the compressor lies in its action on each charge of suction air after it is sealed off from the inlet and before it is brought into communication with the discharge. A charge is initially enclosed in the space bounded by the tooth flanks, casing bore, and end walls. The rotor helices are so chosen that a particular thread space is completely filled and sealed off from the suction just as it is entered at its opposite end by a coacting lobe on the other rotor.[5] Further rotation establishes an axial seal, which separates this charge from the charge in the succeeding grooves. As rotation proceeds, this seal moves axially, effecting a reduction in volume of the charge and a substantially adiabatic compression. When the leading lobes of the grooves pass the boundaries of the discharge port, the compressed medium is brought into communication with the discharge. The location of the port thus determines the "built-in" compression ratio. A typical relationship of charge volume to rotation is illustrated in Fig. 3.

Theoretical indicator cards, assuming no leakage and no dynamic effects, are shown on Fig. 4A for an Elliott-Lysholm compressor operating against various back pressures. The shaded areas represent excess work performed as a result of operating at a pressure ratio different from that "built in." The hatched areas represent work saved by accomplishing adiabatic, as opposed to constant-volume compression. Tests confirm that an accurate location of the discharge ports is not essential. With the slight leakage and the dynamic effects introduced in the design by high peripheral speed, the actual indicator diagrams follow more nearly the type illustrated in Fig. 4B. With high-speed operation, it is thus possible to secure an optimum operating ratio considerably in excess of the "built-in" value.



■ Fig. 4 – Ideal and actual indicator diagrams for compressor with "built-in" pressure ratio of 1.6

Numerous variations in the rotor geometry, associated with lobe height, number of lobes, rotor-speed ratios, and length-to-diameter ratio, have been incorporated in the machines already constructed to suit particular applications. Good space utilization is attained with a few lobes having a large height, whereas better port areas are available when a larger number of lobes are employed. In general, tooth ratios from 1:1 to 1:1.5 have been found most effective.

The tooth profiles are virtually all addendum on the rotor with convex lobes, while the mating rotor has a tooth form largely dedendum, except for a small addendum and

[4] U. S. Letters Patent: 2,243,874; 2,289,371; 2,174,522; 2,111,568.

[5] This choice results in approximately 240-deg wrap of thread on the convex lobed rotor, and yields the largest discharge ports compatible with full utilization of the thread spaces.

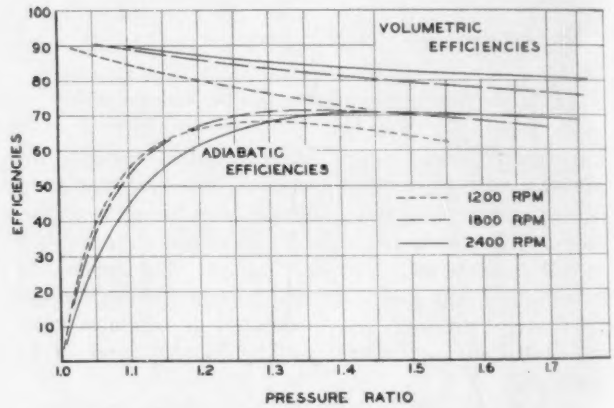
dedendum modification on the respective rotors to replace otherwise sharp corners by radii and fillets. The convex rotor absorbs approximately 110% of the power; the concave rotor being driven by the air. The small torque transmitted by the timing gears maintains the teeth in continuous contact on one side. For certain positions in the rotation of the rotors, small pockets are known to exist at each end which do not communicate with their respective ports. This condition is often relieved by appropriate grooves in the rotors, but on small compressors this refinement is generally unnecessary.

An Elliott-Lysholm machine for supercharger applications having a capacity of from 400 to 500 cfm is shown in Fig. 5. In the development period, wooden models were

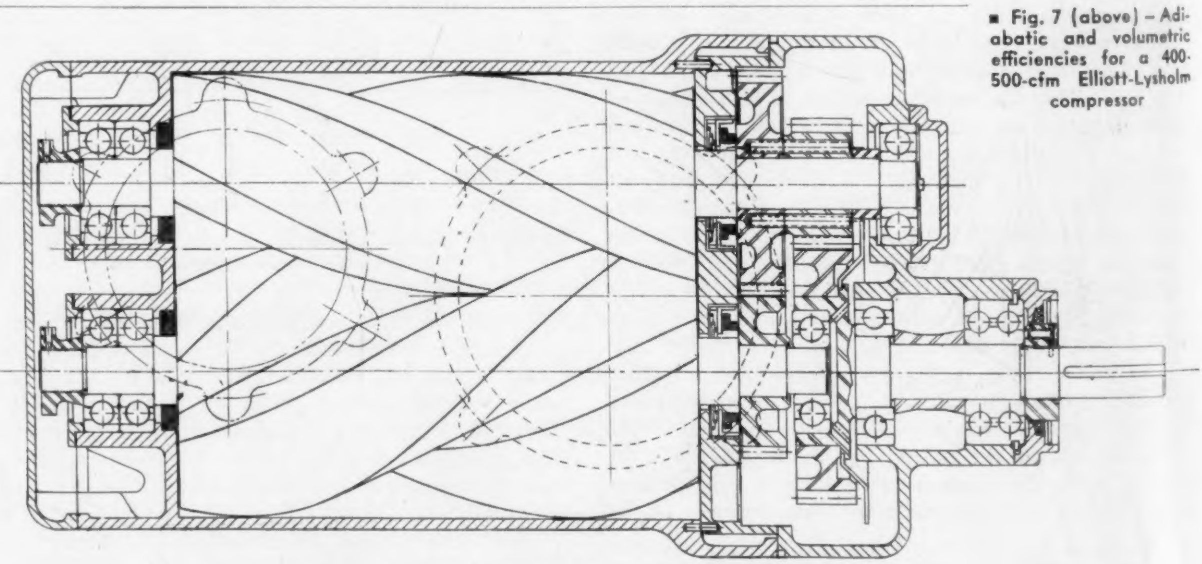
■ Fig. 5 – 400-500-cfm Elliott-Lysholm compressor and model

used as an aid to solution of the design and arrangement problems. Both the final model and the actual machine are shown in the illustration. To eliminate the disadvantages associated with their distortion and high temperature expansion, light metals have not been employed. Completely equipped and constructed of cast iron, the compressor weighs only 80 lb – a feature which can be attributed to the effective use of high rotating speed inherent in the Lysholm arrangement. The compressor is driven through a coupling or by means of a belt at from 2000 to 3000 rpm. Speed increasing gears, shown in the sectional view of Fig. 6, transmitting the driving torque to the timing gears, are incorporated within the casing and are self-lubricated from the oil reservoirs at each end. Hardened and ground airplane-type gears are employed. Leakage of oil into the intake of the machine is eliminated by a carbon nose ring rubbing on the lapped faces of the timing gears, whereas on the outlet end air leakage from within is throttled through close fitting carbon rings. The rotating parts are supported on anti-friction bearings designed for an average life of 50,000 to 100,000 hr. A complete mock-up of the rotating system was operated continuously in the laboratory for nearly 1000 hr at full speed and overload torque before a final design was attempted. The exit porting was arranged for a built-in compression pressure ratio of 1.3.

The adiabatic and volumetric efficiencies of the compressor, measured with a calibrated orifice and with the machine cradled in a dynamometer, are shown in Fig. 7. A maximum efficiency of 72% has been achieved, with performance over much of the speed range in excess of 70% for pressure ratios as high as 1.7. The flatness of the efficiency characteristic with respect to speed and pressure ratio has been exhibited by all Lysholm compressors. It is, in part, due to the fact that the decrease in volumetric loss with speed is compensated by the increased dynamic losses at the inlet and outlet; and in part to the increase in mechanical efficiency as the leakage losses increase with pressure ratio. A flat performance characteristic is advantageous inasmuch as it means that although the compressor

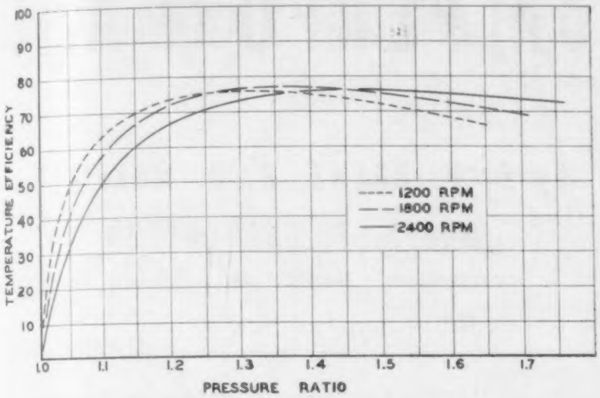


■ Fig. 7 (above) – Adiabatic and volumetric efficiencies for a 400-500-cfm Elliott-Lysholm compressor



■ Fig. 6 – Longitudinal-section of an Elliott-Lysholm compressor

is designed for a fixed compression ratio, its behavior under other than design conditions is so flexible that any mechanical scheme for adjusting the discharge porting is unnecessary. In Fig. 8 is shown the temperature efficiency of the machine as determined from the temperature rise measured with calibrated NACA probes. A maximum efficiency

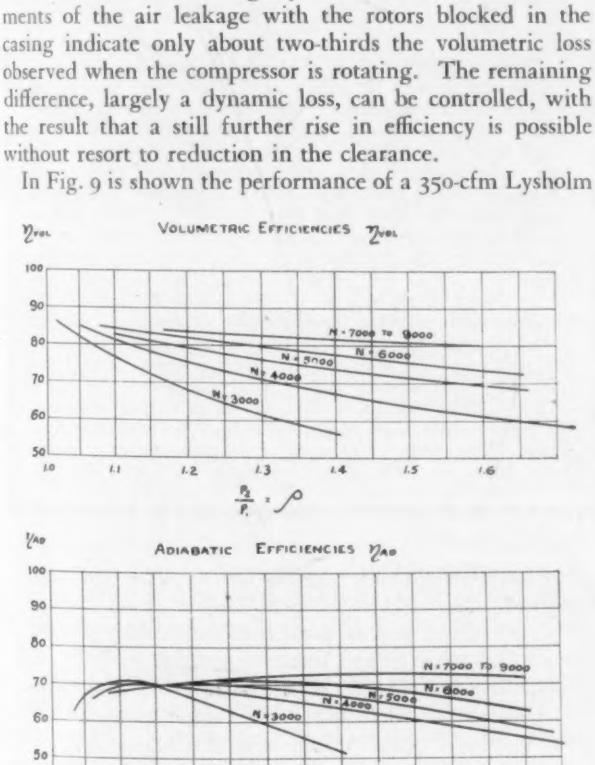


■ Fig. 8 – Temperature efficiency of a 400-500-cfm Elliott-Lysholm compressor

value of 76% has been determined in this manner – the difference between the adiabatic and temperature efficiency being largely the result of 1 hp mechanical loss associated with the bearings, gears, and lubricating means.

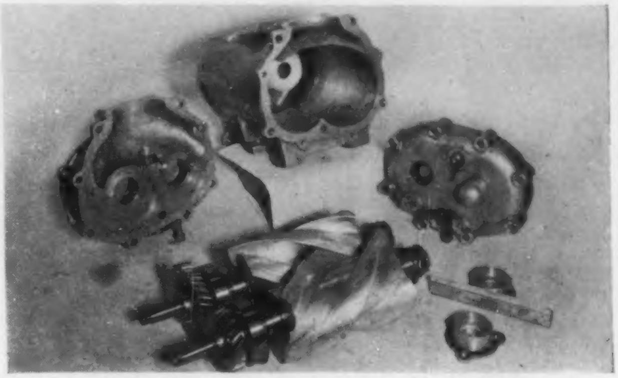
The efficiency attained by this compressor has not been achieved by closing the clearances in the casing and rotor. The design specified 0.005-0.006-in. clearance, which under actual conditions was slightly exceeded. Static measurements of the air leakage with the rotors blocked in the casing indicate only about two-thirds the volumetric loss observed when the compressor is rotating. The remaining difference, largely a dynamic loss, can be controlled, with the result that a still further rise in efficiency is possible without resort to reduction in the clearance.

In Fig. 9 is shown the performance of a 350-cfm Lysholm



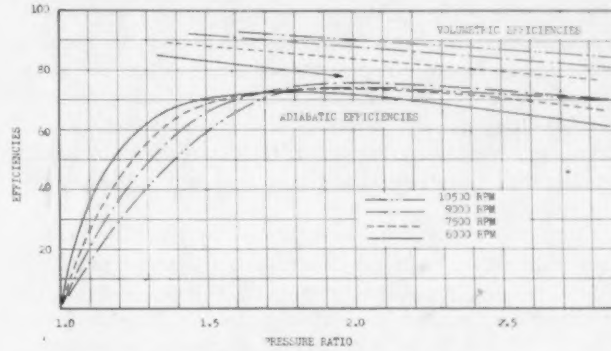
■ Fig. 9 – Adiabatic and volumetric efficiencies for a 350-cfm Lysholm compressor

compressor. The machine was tested on an Elliott dynamometer but is of Swedish design, constructed of light alloys, and weighing approximately 35 lb. Over the active design range the adiabatic efficiency exceeded 71% for pressure ratios between 1.2 and 1.65 and attained a maximum value of approximately 73%. Fig. 10 shows the compressor disassembled.



■ Fig. 10 – Disassembled view of an Elliott-Lysholm compressor

The test results on a 1100-cfm compressor designed for a pressure ratio of 1.5 to 2.0 are shown in Fig. 11. The



■ Fig. 11 – Adiabatic and volumetric efficiencies of an 1100-cfm Lysholm compressor

measurements were made at the DeLaval plant in Sweden in 1937, and resulted in a maximum adiabatic efficiency in excess of 75%. This compressor was designed with a 4-6 tooth combination and a 6-in. male rotor. A later test made after slight alterations to the porting and with reduced clearances resulted in a maximum efficiency in excess of 79% at a pressure ratio of 2.0.

The Lysholm type compressor has already been built in sizes large enough to supercharge the biggest diesel engines that have been designed. For engines operating at constant speed, simple bypass control can be readily incorporated in the design for use at light load, to reduce the friction drag on the engine. For the high pressure supercharging now contemplated, the compressor offers an efficiency exceeding that attained even by the single-stage centrifugal type and without the complication of auxiliary gear transmissions. Two stage units operating at discharge pressures as high as 100 psi have also been built. Light weight units for aircraft service are also feasible and are in fact in the experimental stage of operation abroad.

INFLUENCE of ENGINE ADJUSTMENT
on PERFORMANCE

by D. P. BRENZ, H. H. MAXFIELD, and A. B. CULBERTSON

Shell Oil Co.

GASOLINES of as low as 70 octane number can be used in commercial engines with only slight losses in power and economy, provided proper spark-timing adjustments are made; no major engine changes, such as changes in compression ratios, are necessary.

In order to obtain a complete picture of the relationship between fuel octane number, engine adjustments, and engine performance, the authors carried out a comprehensive investigation.

• • •

THE AUTHORS: D. P. BRENZ has been with the Shell Oil Co. for the past 12 years, four of which were spent in the Engine Research Laboratory as experimental engineer dealing with various antiknock and fuel volatility problems. Since 1935 he has spent his entire time as field engineer, studying the performance of fuels and lubricants under field conditions. Mr. Brenz received his B.S. in mechanical engineering from Kansas State College in 1931. H. H. MAXFIELD was in charge of the engine division of the Sewaren Laboratory of the Shell Oil Co. for five years, and in 1935 became engaged in work on field problems arising from the application of petroleum products. A. B. CULBERTSON received his engineering training at the University of Missouri, and has spent the past 20 years on various technical assignments in the employ of the Shell Oil Co. He has served in such capacities as chief chemist and chief technologist of the Norco Refinery, senior technologist of the Head Office, and since 1940 has been manager of Shell's Products Application Department, where research and study of the performance of petroleum products under service conditions are carried on.

FOR the first time in many years, the steadily increasing quality trend in motor gasoline manufacture has been interrupted by the demands of the war effort, with the result that operators of automotive equipment, since 1941, have been faced with a reduction in antiknock rating of regular-grade gasolines. Prior to the war emergency, regular-grade gasolines had reached a level of 74 to 76 octane number, but late in 1941 this level was reduced to approximately 71 to 72 octane number.

In a large percentage of passenger cars with moderate octane requirements, the reduced octane number caused no change in performance. Other passenger cars, previously knocking or near the knocking point on the higher octane fuels, immediately encountered increased detonation. From the relationships established by previous Cooperative Fuel Research studies on passenger-car engines, it was apparent that retarding the spark timing from 2 to 4 deg would compensate for the reduced octane number of the gasoline without significant losses in power output or fuel economy. In quite a few cases this spark timing correction has been applied, but in most instances the motorist has avoided detonation merely by easing up on the accelerator, which practice will also extend the life of tires and improve gasoline mileage.

From the results of investigations by the Cooperative Fuel Research Committee,* and other investigators, the effects of engine adjustments and octane number on passenger-car engine performance are quite well established,[1, 2, 3] but there is a paucity of similar information on the performance of heavy-duty commercial engines. Because many commercial engines are operated at or near full throttle for a large percentage of their operating time, it was felt that the use of the reduced octane number fuels in commercial engines might involve more serious losses in performance than in passenger-car engines. When encountering objectionable detonation using fuels lower in octane number than those for which the engine was designed, the fleet operator is faced with the problem of making engine readjustments which will reduce detonation and produce the lowest possible losses in power output, gasoline mileage, and engine life.

This investigation was undertaken to provide further information on the fuel requirements of several representative types of modern commercial engines and to evaluate the effects of engine adjustments on the power output and fuel consumption.

From the large fleet of tank trucks available, nine units representing recent-model International, Mack, White, and General Motors Corp. trucks of 2¾- to 4-ton rated capacity were selected for these tests. The major portion of this work was conducted in five units under carefully controlled, heavily loaded conditions in highway operation.

In order to obtain a complete picture of the relationship between fuel octane number, engine adjustments, and engine performance, the following information was obtained:

1. Octane requirement for borderline knock, using standard timing.

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

[1] See SAE Transactions, Vol. 50, October, 1942, pp. 458-464: "1941 CFR Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, W. M. Holaday, and C. B. Veal.

[2] See SAE Transactions, Vol. 34, May, 1939, pp. 210-220: "Spark Timing – Its Relation to Road Octane Numbers and Performance," by L. E. Hebl and T. B. Rendel.

[3] See SAE Transactions, Vol. 33, October, 1938, pp. 427-433: "A Practical Approach to the Road Detonation Problem," by A. J. Blackwood, C. B. Kass, and G. H. B. Davis.

* Now Cooperative Research Council of SAE and API.

and OCTANE NUMBER of COMMERCIAL ENGINES

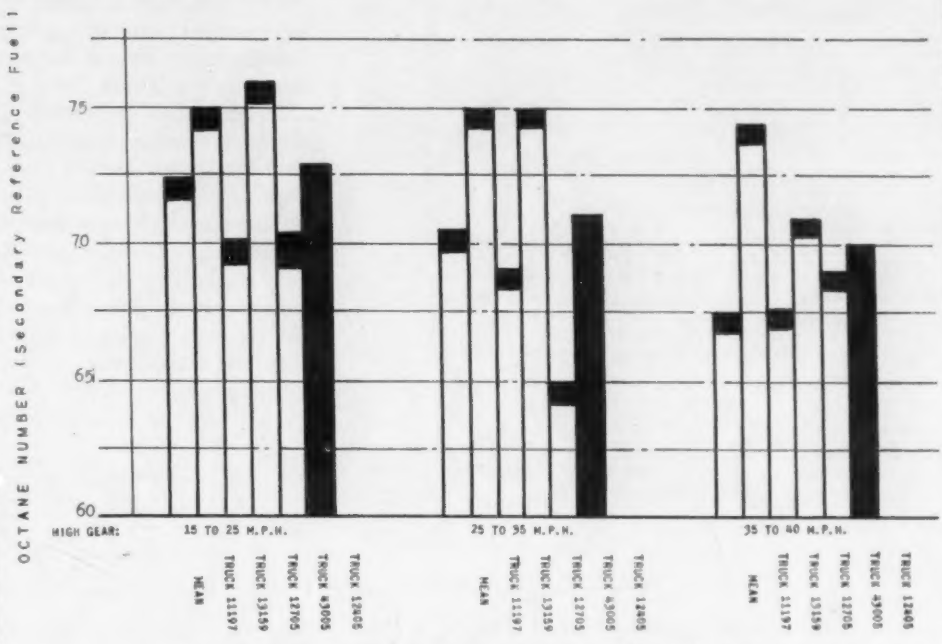
2. Influence of spark timing on octane requirement, power output, and fuel consumption.

3. Influence of carburetion on power output and fuel consumption.

4. Allowable spark timing for various octane number commercial gasolines using normal and over-rich carburetor mixtures.

5. Road antiknock behavior of commercial gasolines of 66, 70.5, 77, and 81.5 octane number.

6. Influence of octane number on actual service operation, as noted by drivers, was investigated by alternately using a 65 octane "third grade" and a 72 octane "regular grade" gasoline in normal transport service.



■ Fig. 1 – Octane requirement for trace knock with standard spark

In conducting this investigation, the "Modified Borderline" knock method was used in determining antiknock performance, as this method appears to give the most complete information on fuel-engine antiknock behavior.[4] Due to the relatively slow acceleration of fully loaded commercial vehicles in comparison with passenger cars, it was not necessary to use extensive electric instrumentation in measuring allowable spark timing. Distributors were equipped with quadrants calibrated in flywheel degrees and manual controls for quickly changing timing and, as the vehicles accelerated in high gear, allowable advances were read directly from the distributor scale. Basic spark timing was checked by means of a neon timing light, and distributor automatic advance on a distributor test stand. Accelerations and mileages were measured by conventional methods, and a Cambridge exhaust gas analyzer was used in checking carburetor mixtures.

In these tests it soon became apparent that the commercial engines tested could readily be adjusted to use the current 71 to 72 octane number gasoline without serious losses in power output or fuel economy. However, of the various methods which can be used to reduce the octane requirement of engines, retarding the spark timing appears the least harmful and is the only readjustment which appears to be necessary for satisfactory operation on the regular-grade gasolines currently supplied.

■ Octane Requirement for Borderline Knock

Periodic surveys by the API and CFR have kept the automotive and petroleum industries well informed as to the octane requirements of passenger cars,[1, 2, 3, 5] but unfortunately little similar information is available on the requirements of commercial engines. For such information to be of any practical significance it should represent the average of a large number of representative units, such as is obtained through the CFR cooperative surveys. Although in these tests octane requirements of only five engines were obtained, the spread in their requirements, which ranged from 70 to 76 octane number under the most severe operating conditions, that is, 15 to 25 mph, indicated that detonation will be encountered with certain heavy-duty engines on the present 71-72 octane regular-grade gasolines unless compensating adjustments are made.

Octane requirements of the engines tested are shown by Fig. 1, and represent the octane number of the secondary reference fuel blend, which just gives borderline detonation when accelerating in high gear at each of the speeds designated on the chart. It is of interest to note that the maximum mean octane requirement was 73, which is compar-

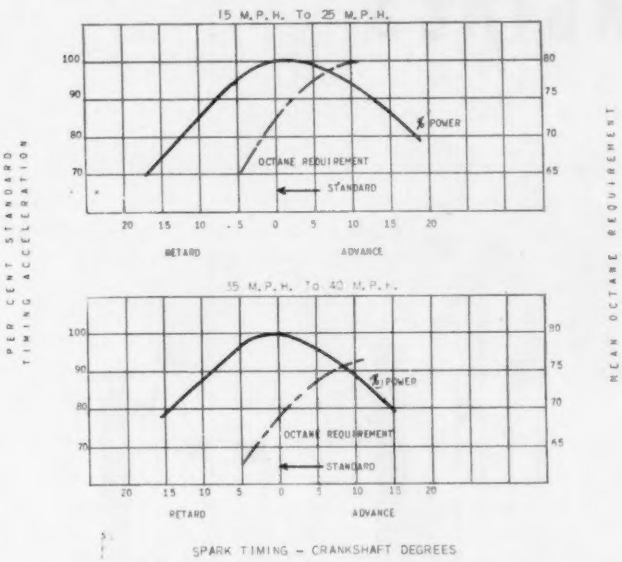
[4] See SAE Transactions, Vol. 33, October, 1938, pp. 436-440: "Spark Advance and Octane Number – A Road-Test Technique," by W. E. Drinkard and J. B. Macauley, Jr.

[5] See SAE Transactions, Vol. 36, May, 1941, pp. 193-204: "1940 Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, and W. M. Holaday.

able with that of passenger cars as determined in recent CFR surveys.

■ Influence of Spark Timing

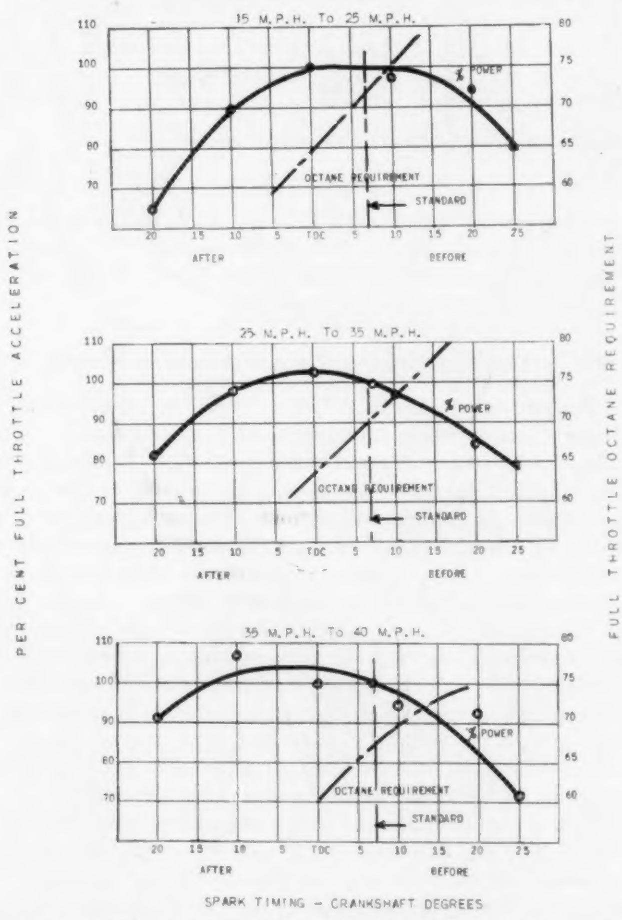
In passenger-car design, relatively high compression ratios are used to obtain good economy, but because of detonation limitations it is necessary, as a compromise, to retard spark timing at full throttle from the maximum power setting. From the results of these tests using heavy-duty engines, it appears that standard spark timing, as specified by the manufacturer, closely approaches full-throttle maximum power timing.



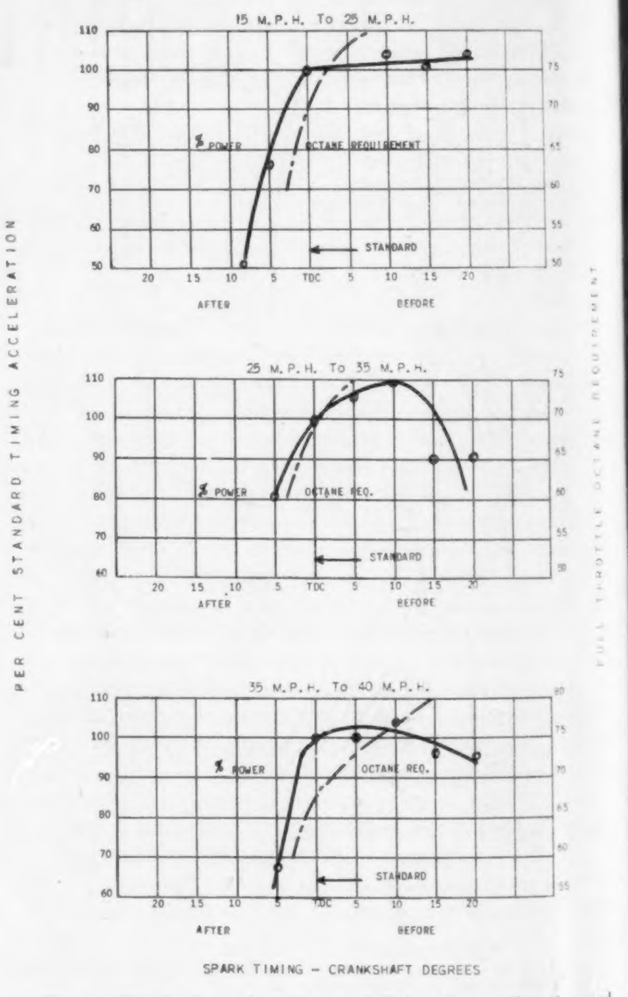
■ **Fig. 2 – Effect of spark timing on full-throttle acceleration and octane requirement – average of tests on trucks Nos. 12405, 12705, 43005, 13159, and 11197**

Usually power measurements are made in the laboratory by an engine block test or on a chassis dynamometer, but to facilitate these tests the time required to accelerate in high gear with full throttle through three speed ranges, for example, 15 to 25, 25 to 35, and 35 to 40 mph, was used as a measure of power output. It has been found that this method correlates satisfactorily with laboratory controlled tests and has the advantage of ensuring actual operating conditions.

Fig. 2 shows the average loss in power and change in octane requirement resulting from changes in spark timing for five engines. These data indicate that retarding spark timing 5 deg from the standard setting gives a 5% loss in acceleration with a corresponding 6 to 8 unit reduction in octane requirement. Individual performance curves for each of the five engines (refer to Figs. 3 to 7 inclusive) indicate only slight losses in acceleration resulted from retarding spark timing to compensate for a 5 point reduction in octane requirement. It will be noted that engines Nos. 12705 and 43005 were relatively sensitive to changes in spark timing or octane number and were the only units



■ **Fig. 3 – Effect of spark timing on full-throttle acceleration and octane requirement – truck No. 12405**



■ **Fig. 4 – Effect of spark timing on full-throttle acceleration and octane requirement – truck No. 12705**

showing an improvement in performance with advanced timing. This indicates that the basic spark timing is slightly retarded from that for maximum power.

It has been shown that small changes in spark timing quite effectively reduced the octane requirement of the engines tested without appreciably affecting power output. Tests were then conducted to determine to what extent such changes would affect fuel consumption.

Mileage tests with fully loaded trucks at 30 mph in high gear showed that retarding spark timing 5 deg gave an average decrease of 2½% in mileage, with a maximum loss of 5% from that obtained at standard timing. This retard in timing reduced the average octane requirement 8 and 6 octane units at low and high speed respectively.

Fig. 8 shows the influence of spark adjustment on average relative mileage as determined for five engines. Specific mileage data for each of the engines are shown by Figs. 9 to 12 inclusive. A comparison of these data with comparable acceleration data indicates that changes in spark timing had less influence on fuel consumption than on power output.

■ Effect of Carburetion on Power Output

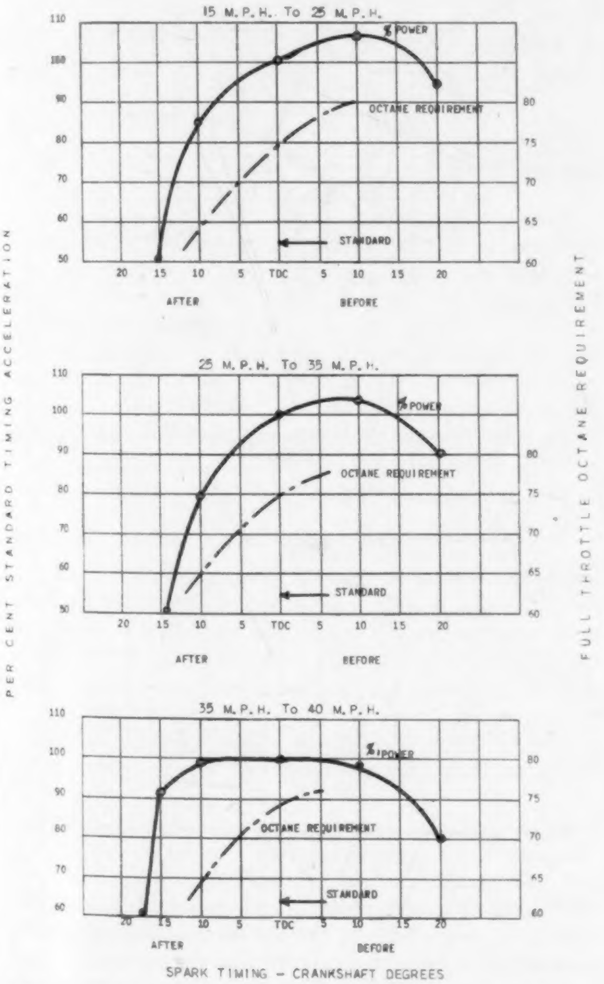
Tests similar to those conducted in investigating the effect of spark timing were made on truck No. 13159 to determine the effect of mixture strength on acceleration. The carburetor main metering jet, which was calibrated to give approximately 13:1 air-fuel ratio mixture at full throttle,[6] was replaced with an adjustable jet and variations in mixture measured with a Cambridge exhaust gas analyzer. Full-throttle, high-gear, acceleration tests were then made at different carburetor settings.

Results of these tests (shown in Fig. 13) indicated maximum power was obtained using the standard full-throttle mixture (13:1 air-fuel ratio). It will be noted that use of either leaner or richer mixtures gave a decrease in full-throttle acceleration.

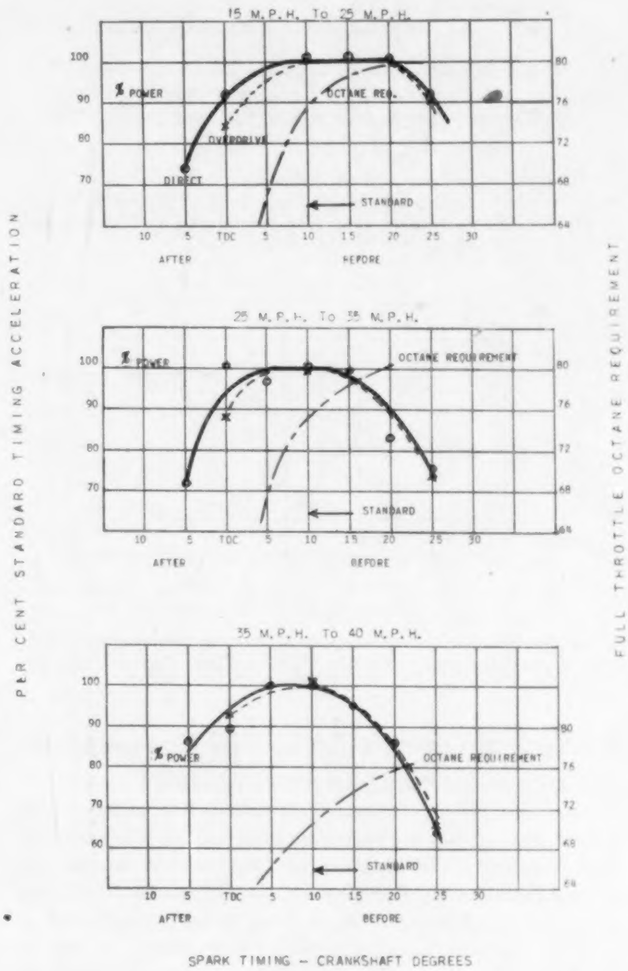
Optimum mixtures for borderline knock operation on three commercial gasolines (66.0, 70.5, and 78.5 octane number) using standard spark are also shown. The data indicate that knockless operation on the 70.5 octane fuel required enriching the mixture from 13:1 to 9.6 air-fuel ratio for a subsequent loss in acceleration of 13% at low speed and 10% at high speed.

This relatively large decrease in power, resulting from enriching the mixture to compensate for detonation on the 70.5 octane fuel, is in contrast to a 3% loss which resulted from retarding the spark 6½ deg with no change in carburetion.

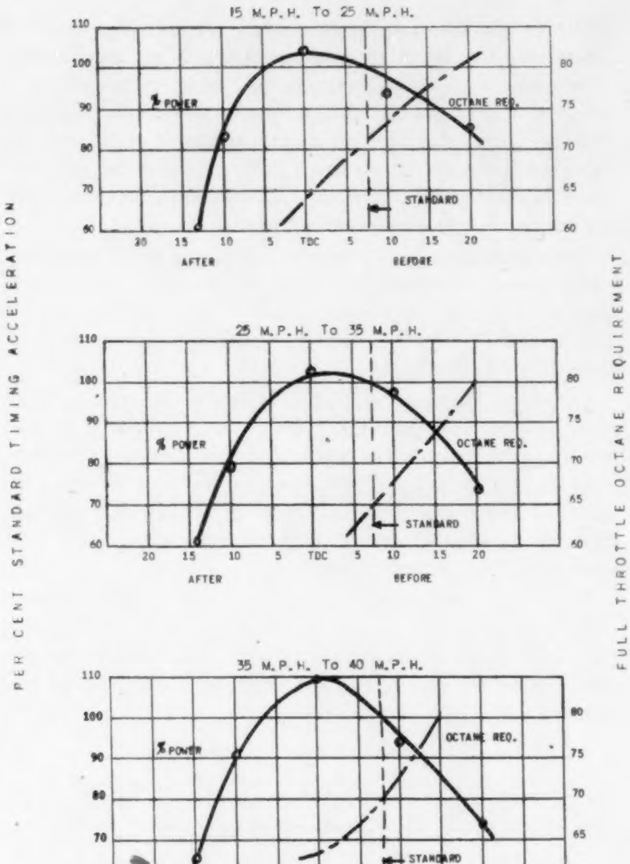
[6] See SAE Transactions, Vol. 36, April, 1941, pp. 160-164: "A 13-year Improvement in Mixture Ratios," by W. G. Lovell, J. M. Campbell, B. A. D'Alleva, and P. K. Winter.



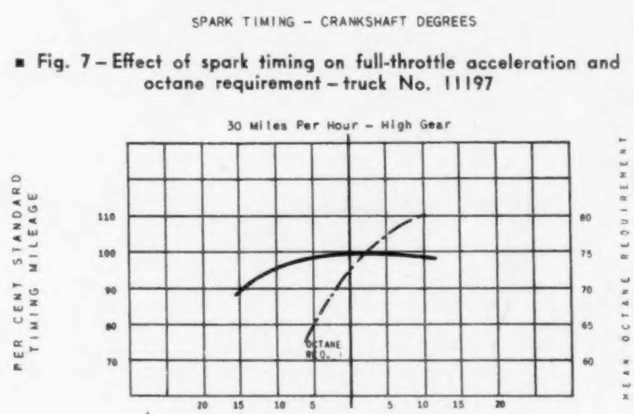
■ Fig. 5 – Effect of spark timing on full-throttle acceleration and octane requirement – truck No. 43005



■ Fig. 6 – Effect of spark timing on full-throttle acceleration and octane requirement – truck No. 13159



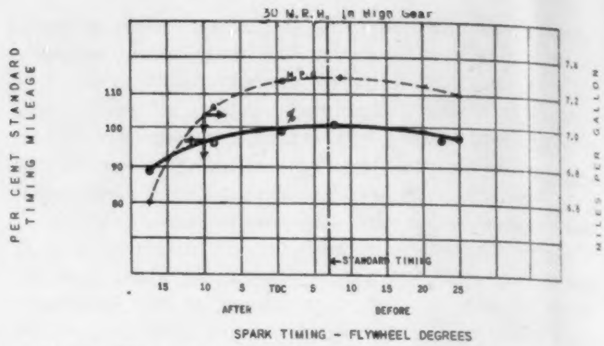
■ Fig. 7 – Effect of spark timing on full-throttle acceleration and octane requirement – truck No. 11197



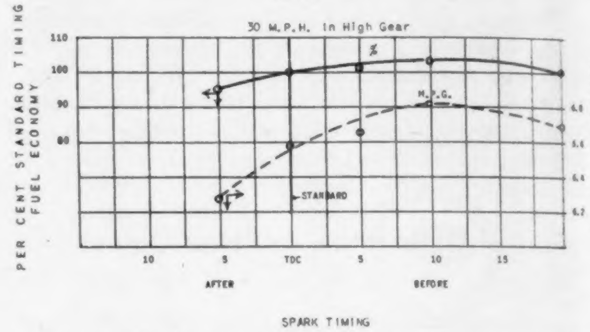
■ Fig. 8 – Effect of spark adjustment on fuel consumption – average of road tests on trucks Nos. 12405, 12705, 43005, 13159, and 11197

■ Effect of Carburetion on Fuel Consumption

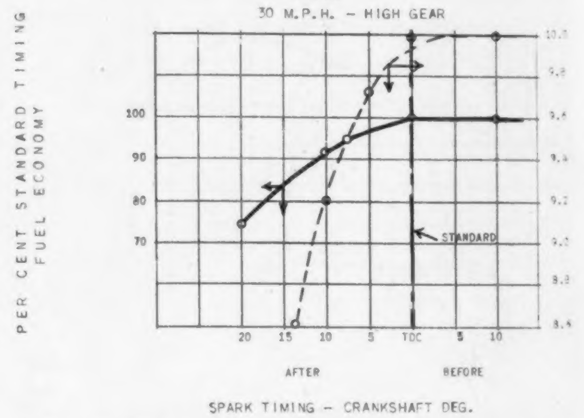
In addition to investigating the influence of mixture on acceleration, mileage tests were made in the same vehicle (truck No. 13159) to determine the effect of this factor on fuel economy. The exhaust gas analyzer indicated that normal carburetor metering for this engine when operating on a level highway, in high gear, fully loaded, and at 30 mph was 13.7 air-fuel ratio. While extensive tests to determine the effect of carburetion on economy were not made, results shown in Fig. 14 indicate a 15% decrease in economy by changing from 13.7 to 11.7 air-fuel ratio.



■ Fig. 9 – Effect of spark timing on fuel consumption – truck No. 12405



■ Fig. 10 – Effect of spark timing on fuel economy – truck No. 12705



■ Fig. 11 – Effect of spark timing on fuel economy – truck No. 43005

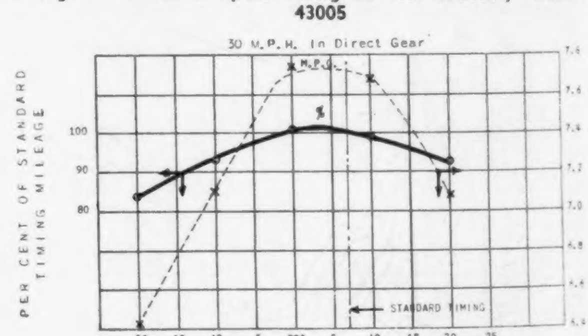
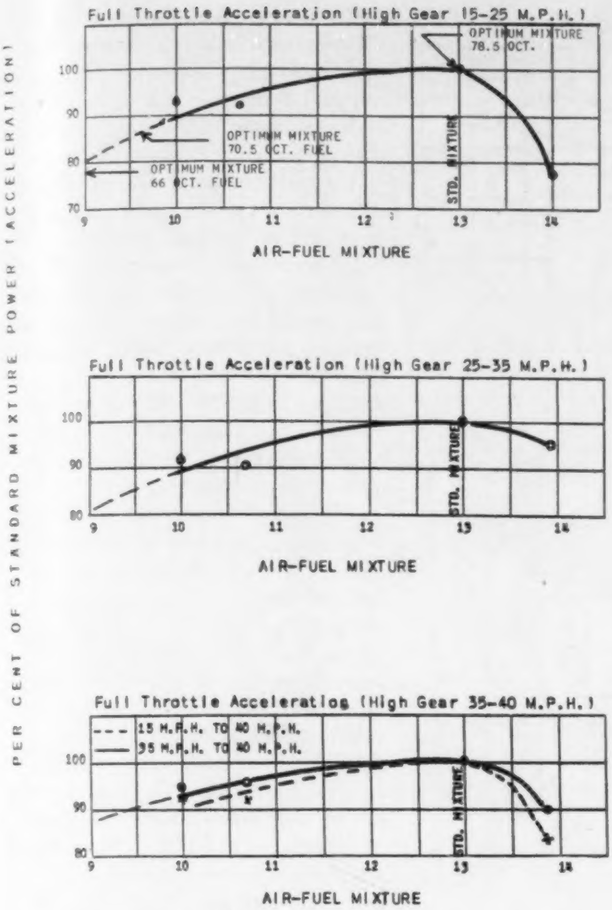


Fig. 12 – Effect of spark timing on fuel consumption – truck No. 11197

On the two curves showing the effect of mixture and timing on fuel economy, the optimum settings for borderline knock of 66.0, 70.5, and 78.5 octane commercial gasolines are indicated. It will be noted that knockless performance was obtained on the 70.5 octane fuel by either



▪ Fig. 13 – Effect of carburetor mixture on full-throttle acceleration – truck No. 13159

enriching the carburetor mixture 3.4 air-fuel ratios or by retarding timing 5 deg. Enriching the mixture gave a mileage loss of 25%, but retarding spark timing decreased mileage only about 4%.

From the results of these tests, it does not appear feasible to lower engine octane requirements by enriching carburetion. Such compensating adjustments appear to give much greater relative loss in power and economy, for a given octane reduction, than is obtained from retarding spark timing.

▪ Spark Timing for Various Gasolines

It has been mentioned that the average octane requirement of the five engines tested under the most severe conditions was 73 octane number, based on equivalent reference fuel octane numbers. However, subsequent tests indicated that, due to the severity of the heavy-duty engines when operating under full load, an estimated 75 ASTM octane commercial gasoline would be required for borderline knock operation when using standard timing and carburetion.

Allowable spark settings, as determined for fuels A, B, C, and D, are shown by Figs. 15 and 16. These data indicate that use of the 66 octane fuel necessitated retarding the timing an average of 9 deg, while the 77 octane fuel permitted advancing the timing 1½ deg. It will be noted that allowable spark settings varied considerably between the engines tested depending upon their severity and octane requirement; that is, using the 70.5 octane fuel, optimum settings varied from ½ deg advance to 8 deg retard or an average of 4½ deg retard.

Data in Figs. 15 and 17 show the effect of variations in air-fuel mixture on allowable timing for truck No. 13159 and indicate that fuel A (66 octane) could be burned without knocking either by using normal carburetion and retarding the spark 10 deg or by using standard timing and enriching the carburetor mixture (9:1 air-fuel ratio).

Having determined the effect of spark timing and carburetion on power output and economy, it is possible, by knowing the allowable timing (for borderline knock) to predict the relative performance of various commercial gasolines. For example, it will be noted from the data in Fig. 15 that fuel C (77 octane) had an average allowable timing of 1½ deg advance in comparison with 4½ deg retard for fuel B (70.5 octane).

Then referring to the power loss (decrease in acceleration) curve in Fig. 2, it will be noted that:

1. Maximum acceleration was obtained on fuel C at an optimum spark setting of 1½ deg advance.

2. A 3 to 5% power loss was obtained on fuel B using allowable timing of 4½ deg retard.

Referring to Fig. 8, it will be noted that:

1. Maximum economy was obtained on fuel C.

2. Approximately a 2% loss in fuel economy was obtained from the use of fuel B.

▪ Road Antiknock Performance

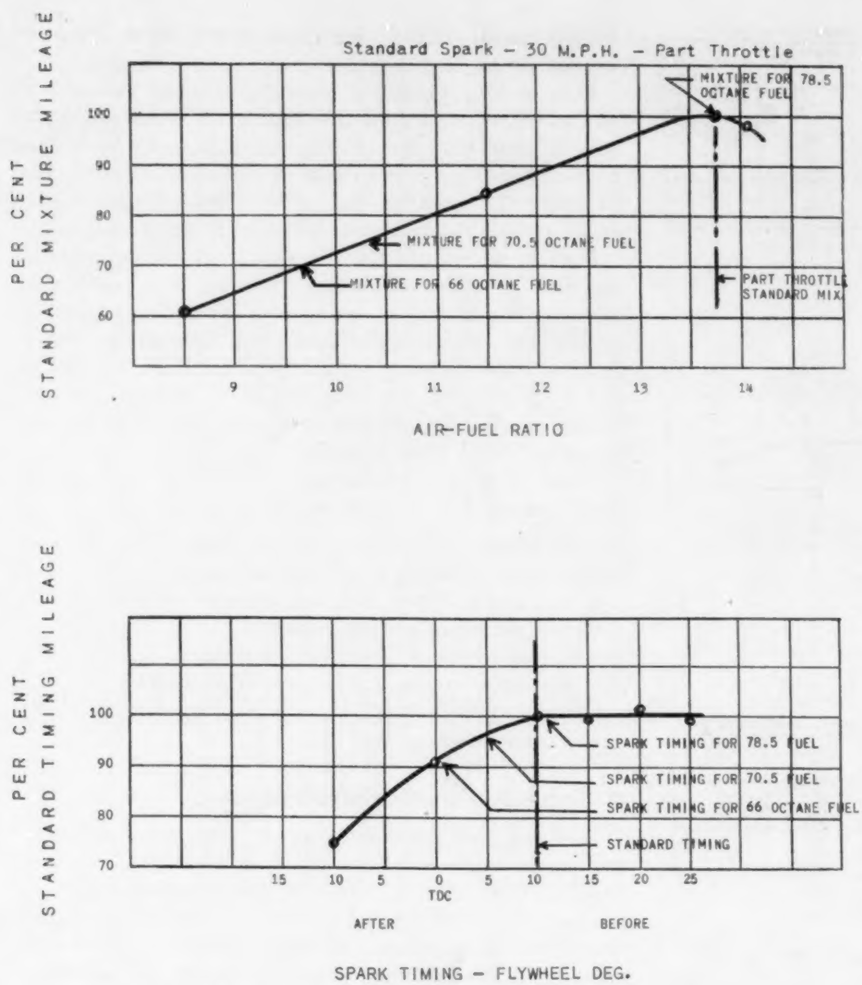
To make a complete study of the fuel requirements of commercial engines, it is essential to investigate the antiknock performance of commercial types of gasolines in actual service. Therefore, in addition to determining engine octane requirements as previously discussed, antiknock behavior of the following gasolines, having normal sensitivities (difference between Research and ASTM ratings), was investigated by the "Modified Borderline Knock Method."

Octane Ratings of Commercial-Type Gasolines Tested

	Fuel A	Fuel B	Fuel C	Fuel D	Fuel E*
ASTM Octane No.	66.0	70.5	77.0	81.5	78.5
Research Octane No.	68.0	75.5	82.0	87.0	86.5
Sensitivity	2.0	5.0	5.0	5.5	8.0

* Units Nos. 11197 and 13159.

By this method, which is generally recognized as the test procedure giving the most accurate picture of engine and fuel antiknock performance, allowable spark timing giving trace knock at full throttle was determined for test fuels as well as secondary reference fuel blends throughout the speed range. Then by comparing allowable spark timing for each of the test fuels with that of reference blends, it is possible to determine the equivalent road octane rating of the fuels at any speed. The tests were made on level highway in high gear with units fully loaded. Under these conditions, the acceleration was so slow that it was possible to advance the distributor manually to the borderline knock point, and read the advance from a scale on the distributor at each 5-mph increment. Thus in Figs. 18, 20, 23, and 24, the spark advance values indicated on the reference-fuel framework charts for each of the trucks represent automatic advance plus the manual spark setting. In each case, the distributor advance characteristics are shown with the basic static timing set according to the manufacturer's recommendation. From these curves it is



■ Fig. 14 – Effect of carburetor mixture on fuel consumption (upper curve); effect of spark adjustment on fuel consumption (lower curve) – truck No. 13159

possible to determine the knocking tendency of any fuel in comparison with that of standard reference fuel blends. For instance, data for truck No. 12405, Fig. 18 (upper graph), show that allowable timing using fuel C was comparable at 20 mph with that of a 75 octane reference blend, and at 40 mph with about a 71 octane fuel. These values then represent the road octane rating of the test fuel at each of the speeds referred to. It will also be noted that the octane requirement of this engine, using standard timing, is 72.5. Equivalent road octane ratings of each of the test gasolines, as calculated from the borderline knock curves, are shown in Figs. 19 and 21.

Fig. 22 shows the average road octane ratings of each of the commercial-type gasolines as obtained at various speeds in four engines. It will be noted that these road ratings are appreciably below the octane ratings as determined by the ASTM "Motor" method, which, according to the 1940 CFR tests,[5] correlate reasonably well with passenger-car performance. Since these tests indicate that commercial-type gasolines are appreciably depreciated in heavy-duty commercial engines, it appears that it will be advisable to include this type vehicle in future CFR engine-fuel surveys to assure a complete picture of the antiknock behavior of current commercial gasolines. A further confirmation of the relative severity of commercial engines, in comparison with passenger-car engines, is the fact that it was impossible to rate gasolines by either the "Uniontown" knock intensity or conventional "borderline" knock method because of the tendency for a light knock to build up in intensity to preignition conditions.

Fig. 23 compares the antiknock behavior of fuels A and B, using 9.1 and 13.8 air-fuel ratio mixtures. The data are self-explanatory and show the relatively greater allowable timing possible using rich mixtures in comparison with standard mixture. However, it has been mentioned that due to the relatively large losses in power and economy, this is not a practical method of compensating for reduced octane number.

Allowable spark advance and road ratings, as determined on truck No. 11197, are shown by Figs. 24 and 25, but were not included in computing the average ratings (Fig. 22) because of an obvious abnormal increase in engine severity resulting from a restriction in the cooling water system. The fact that an 81.5 ASTM octane number commercial-type fuel rated only 67 octane number at high speed under this abnormally severe condition is of interest, since it represents a condition likely to be encountered in service. This engine performed normally until subjected to prolonged full-throttle operation, and it was believed, until the time of these tests, that the overheating condition was due to a rebuilt engine.

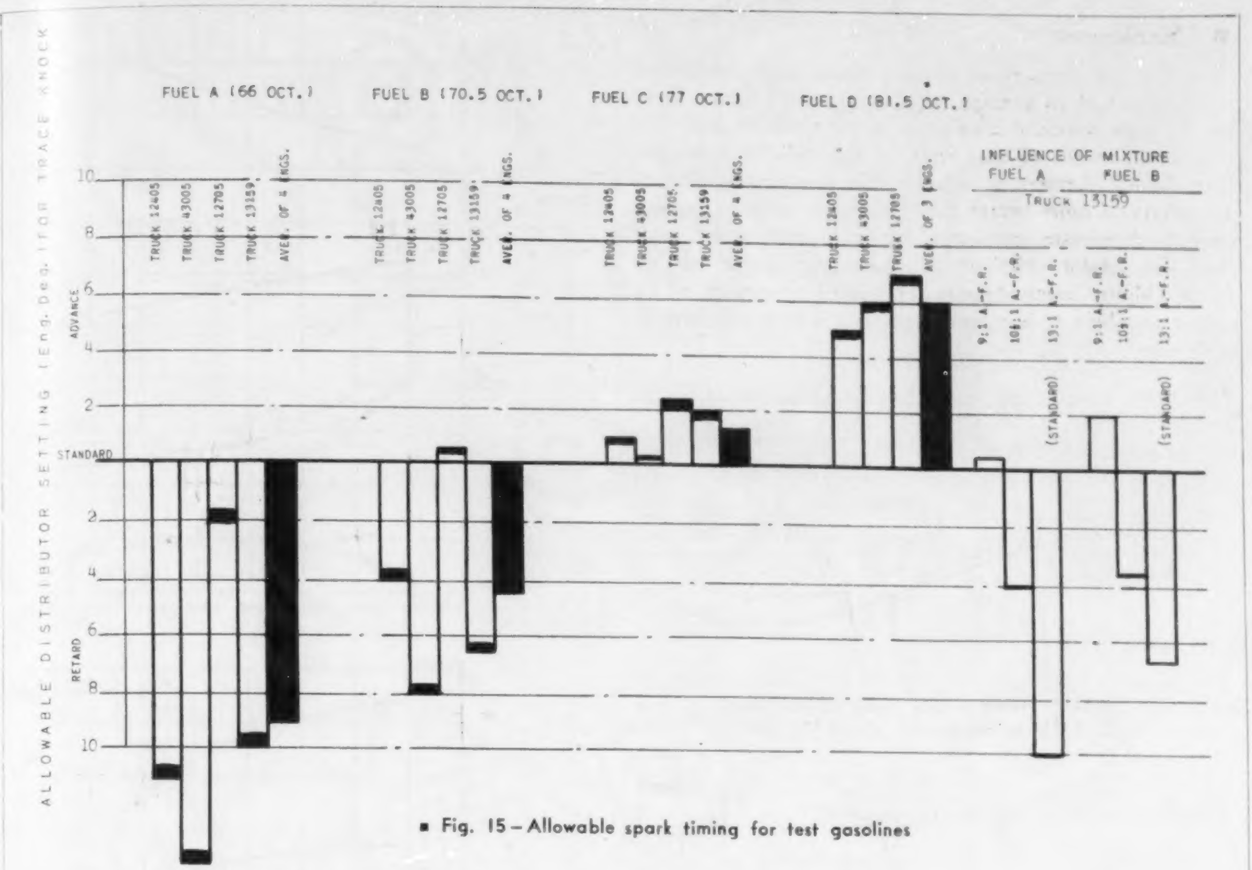
■ Detecting Changes in Octane Number

Subsequent to the foregoing tests, conducted under controlled road test conditions, tests were conducted, using heavy-duty transport trucks in regular service, to determine the influence of changes in octane number on overall performance as noted by the drivers. Five tractor-trailer transport units were each operated from 1200 to 2500 miles on two currently available commercial gasolines having the following antiknock properties:

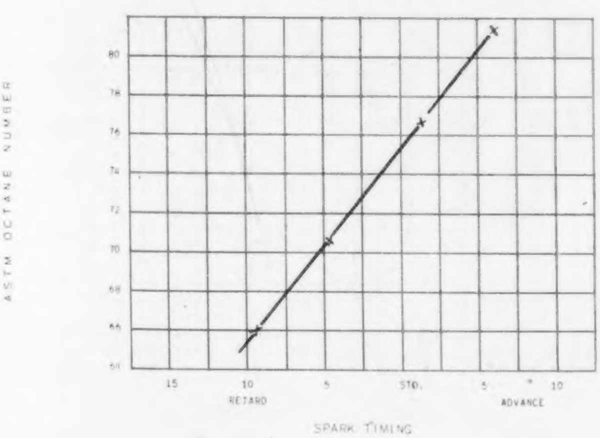
	Third Grade	Regular Grade
ASTM Octane No.	65.0	72.0
Research Octane No.	70.5	78.0
Tel, cc per gal	0.16	0.16

In connection with these service tests, each driver was required to submit a daily report on the performance of his vehicle, and it is rather interesting to note that none of the drivers detected a difference in performance between the 65 and 72 octane number gasolines, although results of tests under controlled conditions indicated that about 5% loss in power could be expected.

Although results of these tests indicated satisfactory performance from a fuel of relatively low antiknock quality, the following two points should be considered before concluding that such a fuel would be fully satisfactory in service:



■ Fig. 15 – Allowable spark timing for test gasolines



■ Fig. 16 – Average allowable spark timing for commercial gasolines – average of units 12405, 43005, 12705, and 13159

1. The optimum spark timing for obtaining maximum performance from a given fuel will vary, even among engines of the same model, depending on engine adjustments and deposits. Therefore, to use a fuel with a slight deficiency in antiknock quality would probably necessitate determination of the allowable timing for each individual engine in a fleet, or at least a close check on the estimated spark adjustment.

2. Use of a fuel having antiknock quality in excess of that required for obtaining maximum performance allows a factor of safety for maladjustments and abnormally severe operating conditions. In one case a relatively low octane fuel might show entirely satisfactory results, while in another the same fuel might give unsatisfactory results even in the same type of equipment.

From the foregoing, it is evident that although satisfactory performance can be obtained from a 65 octane fuel, the use of such a fuel would probably require somewhat closer maintenance than would be necessary if a higher octane fuel were used.

■ Survey of Driving Habits

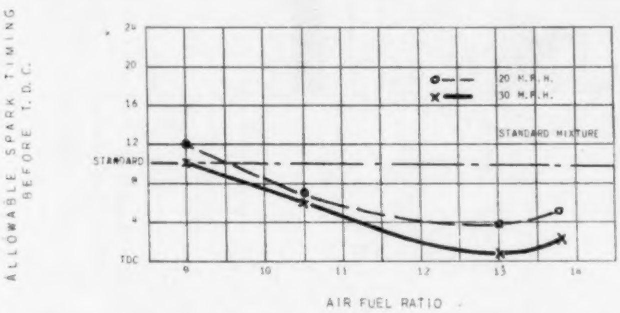
A most interesting observation was that a number of the drivers "lugged" the engines rather than shift to a lower gear maintaining a reasonable engine speed. The ease with which the gears could be shifted seemed to influence the drivers' reaction to proper shifting. It was noted that the drivers invariably "lugged" the engines in trucks which were difficult to shift. Since this low engine speed condition, with full throttle, represents the condition at which the highest octane requirement is obtained, it appears that this condition (15 to 25 mph in high gear) will determine the octane requirement of the engine in service. Further, the low-speed antiknock performance of fuels should be considered as well as the high-speed performance because of such operating conditions in service.

Due to the heavy gross loads (approximately 22,000 lb) during these tests the observers estimated that full throttle was used over half of the time. This demonstrates the severe operating conditions to which commercial truck engines are subjected in service in comparison with passenger-car engines, which operate only a small percentage of the time under full-throttle conditions.

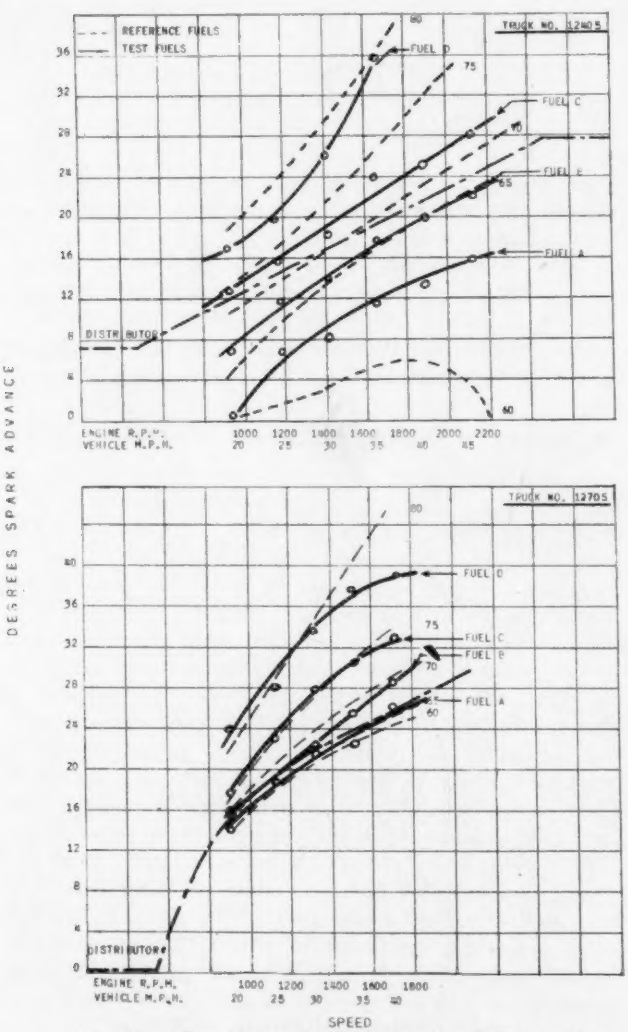
■ Conclusions

1. The five commercial engines tested, under controlled conditions, had an average requirement of 73 octane number for trace knock at maximum power spark timing.

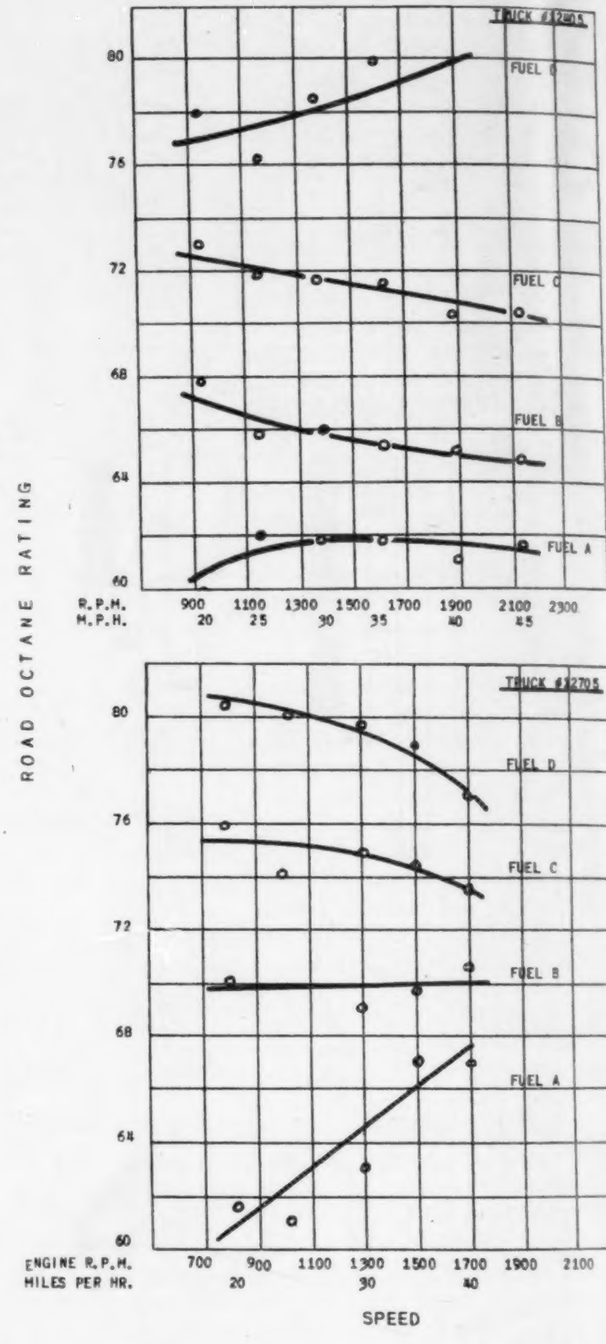
2. From results of road tests, by the "Modified Borderline" knock method, it appears that commercial engines are relatively more severe than passenger-car engines, and tend to depreciate commercial fuels in comparison with the CFR "Motor" Method. A typical 70.5 octane number (CFR "Motor" method) was depreciated an average of 2.5 octane numbers at low speed, and 4.0 octane numbers at high speed.



■ Fig. 17 – Allowable spark timing using various mixtures – fuel A – 66 ASTM octane number – truck No. 13159



■ Fig. 18 – Allowable spark advance for borderline knock

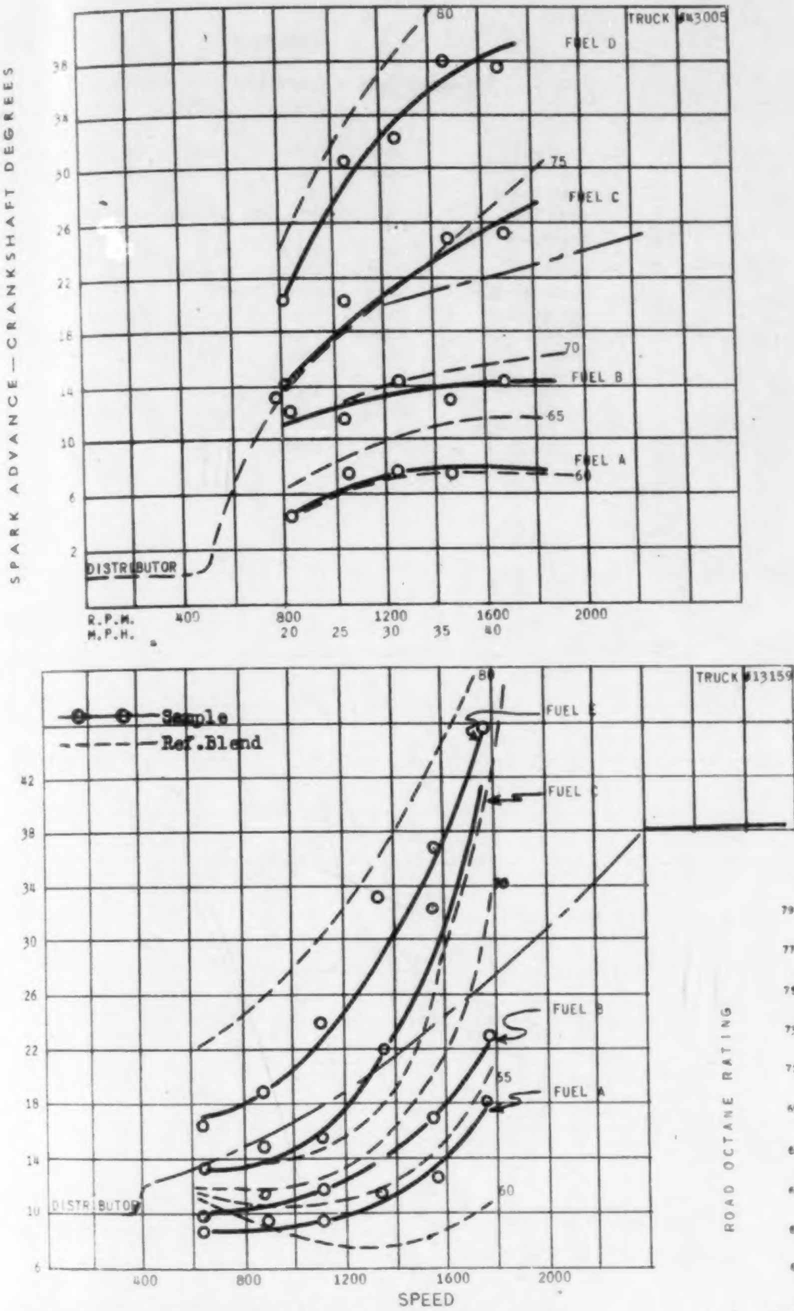


■ Fig. 19 – Road octane ratings of commercial gasolines

3. Retarding spark timing was indicated as the most satisfactory method to compensate for reductions in octane number. Results of road tests, under controlled conditions, showed average allowable timing of 1½ deg advance (from standard), 4½ deg retard, and 9 deg retard for trace knock operation on three representative commercial gasolines of 77.0, 70.5, and 66.0 ASTM octane number respectively. The test data indicate that when using optimum spark settings for each fuel, a loss of about 4% in full-throttle acceleration and 1% in fuel economy would result from using the 70.5 octane number in comparison with the 77.0 octane fuel.

4. Performance of the commercial engines tested ap-

SPARK ADVANCE — CRANKSHAFT DEGREES



■ Fig. 20 – Allowable spark advance for borderline knock

peared to differ from previously noted passenger-car engine performance in that:

(a) Light to medium detonation can normally be tolerated in passenger-car service, whereas in commercial operation a light knock will increase to preignition conditions under prolonged full-throttle operation.

(b) In the engines tested, no appreciable improvement in power or economy resulted from advancing spark timing from the basic timing specified by the manufacturers, but in passenger-car engines the basic timing is often retarded 5 to 10 deg from the maximum power setting because of detonation requirements. This indicates that the use of fuels having octane ratings above the octane requirement (at basic setting) of the engine may give no improvement in performance in commercial engines.

5. The recent drop in regular-grade gasolines, from 75.0-76.0 to 71.0-72.0 ASTM octane number, is indicated from these tests to have no important effect on fuel economy or power output in heavy-duty service when spark timing is properly retarded for the lower octane number fuel.

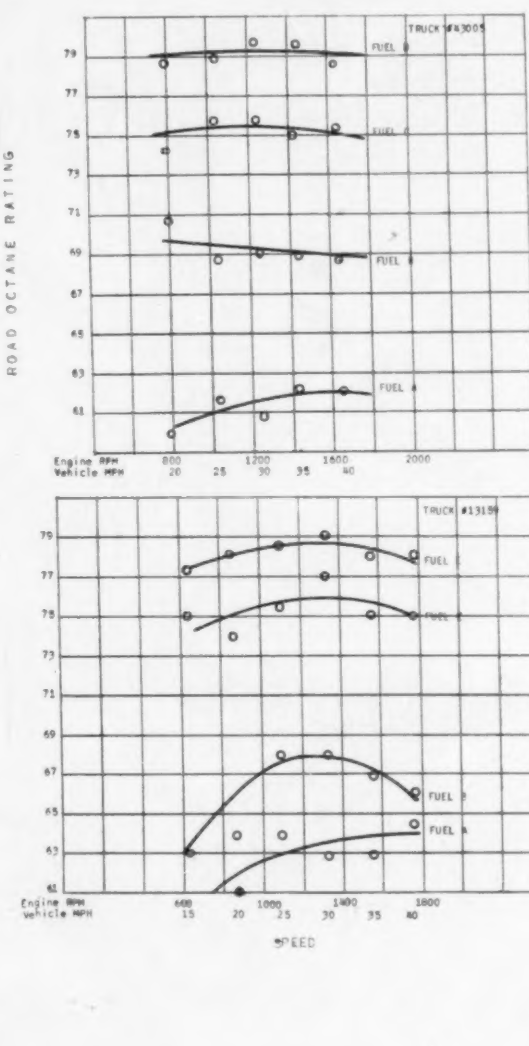
6. No major engine changes, such as changes in compression ratios, are necessary unless fuels of lower than 65 octane number are used.

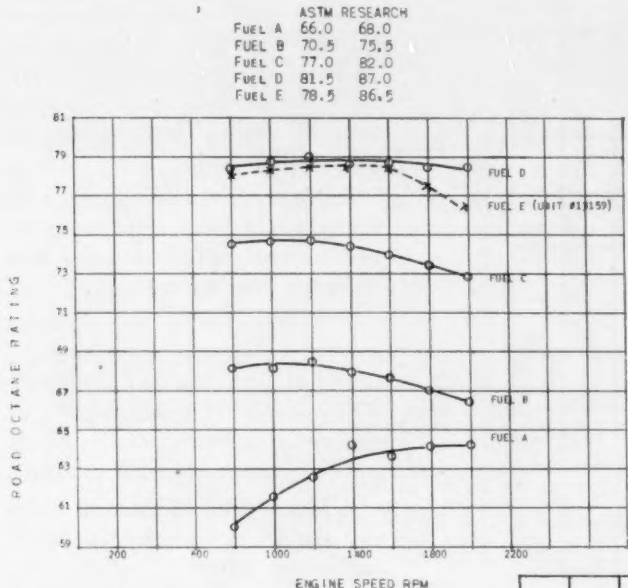
7. Service tests in five transport trucks in intra-city and inter-city operation indicated no significant differences in fuel consumption or power output, when using 65 octane and 72 octane commercial gasolines.

8. Heavy-duty commercial vehicles should be included in future engine-fuel surveys.

continued on page 208

■ Fig. 21 – Road octane ratings of commercial gasolines



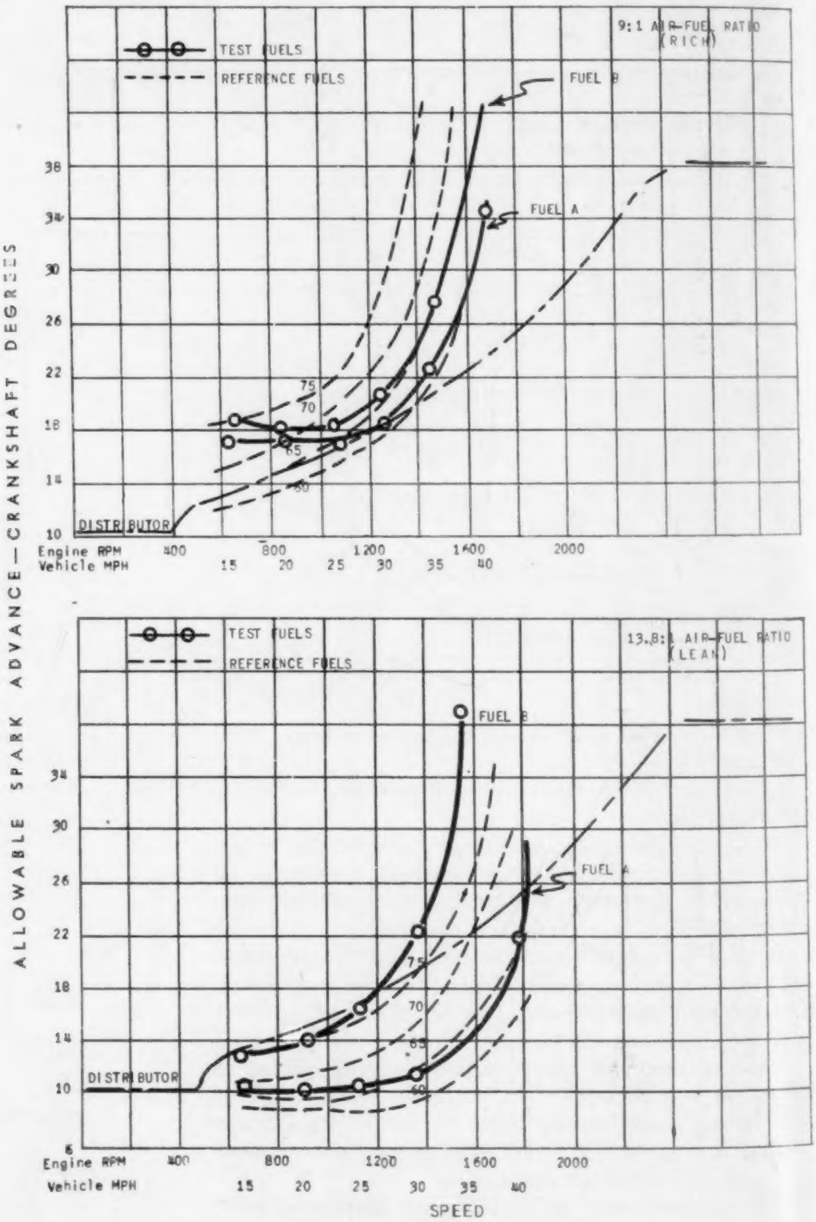


■ Fig. 22 – Mean road octane rating of test fuels in four commercial engines -- trucks Nos. 12405, 12705, 43005, and 13159

Appendix I

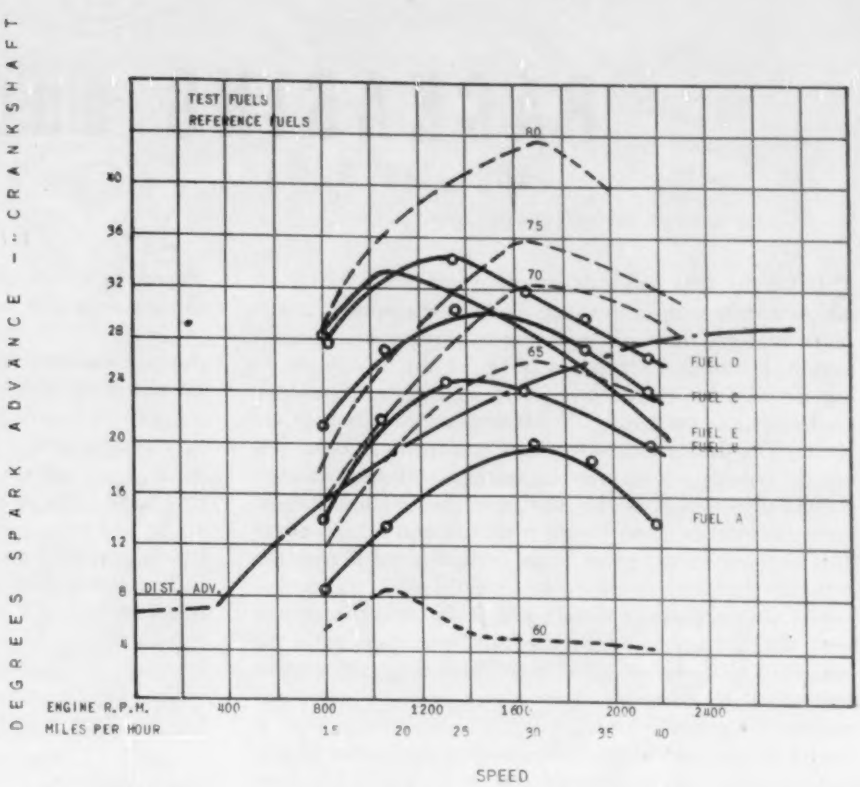
Statistical Data of Commercial Engines Tested (Controlled Test Conditions)

Truck No.	12405	12705	43005	13159	11107
Year	1940	1940	1940	1940	1939
Rated Size, tons	3-4	3-4	3	3	$2\frac{3}{4}$
Rated Bhp	116 at 3000	122 at 2800	110 at 2800	93.7 at 2800	103 at 260
Engine Displacement, cu in.	362	360.8	308.2	298.2	353.8
Bore and Stroke, in.	$3\frac{7}{8}$x$5\frac{1}{8}$	$4\frac{1}{8}$x$4\frac{1}{2}$	$3\frac{13}{16}$x$4\frac{1}{2}$	$3\frac{3}{4}$x$4\frac{1}{2}$	$3\frac{7}{8}$x5
Compression Ratio	5.81:1	6.0:1	6.0:1	5.7:1	5.25:1
Rear Axle Ratio	7.14:1	6.43:1	6.43:1	6.43:1	6.31:1
Overdrive Ratio	0.788:1	0.77:1	0.77:1	0.823:1	0.794:1
Tire Size	10x20	10x20	10x20	9x20	9x20
Gross Load During Tests, lb	22,500	22,000	12,500	9,700	7,500
Total Miles	100,600	67,000	10,000	41,000	50,108
Miles Since Carbon Removed	45,000	35,000	10,000	....	2,000



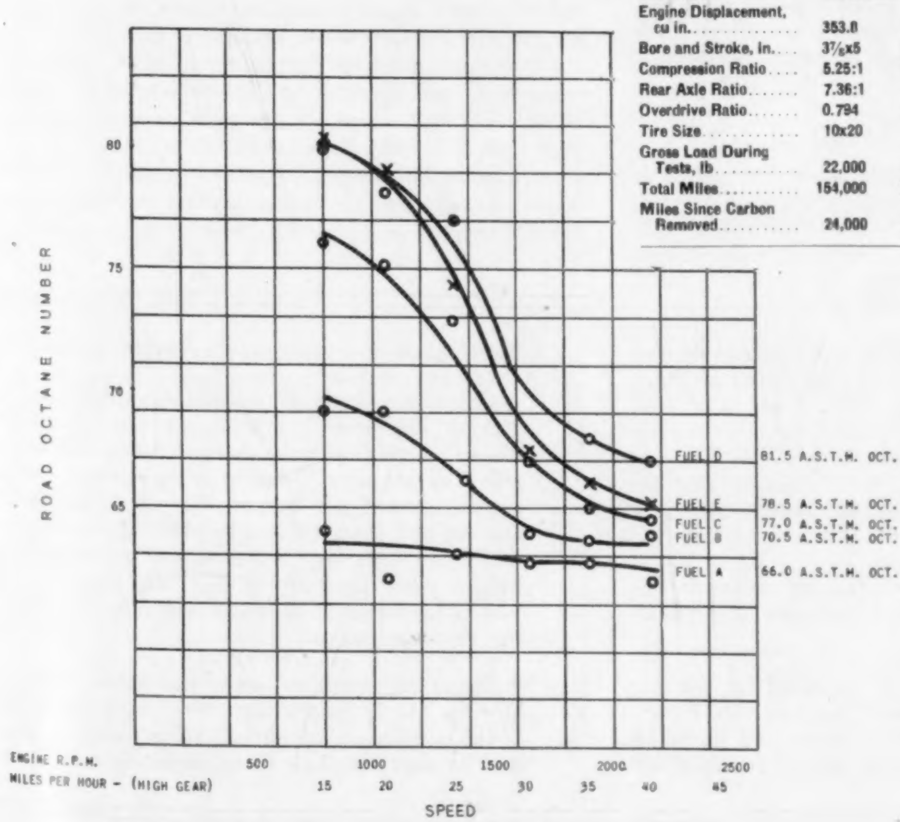
■ Fig. 23 – Curves showing the effect of carburetor mixture on allowable spark advance

■ Fig. 24 – Allowable spark advance for borderline knock in engine with restricted cooling system – truck No. 11197



Appendix 2
Transport Trucks Used in Service Tests

Truck No.	12639	12930	13149	13292	12406
Year	1939	1939	1939	1939	1940
Gross Load During Tests, lb	22,000	22,000	22,000	22,000	22,000
Rated Size, tons	3-4	3-4	3-4	3-4	3-4
Rated Bhp	103 at 2800	116 at 3000	116 at 3000	116 at 3000	116 at 3000
Engine Displacement, cu in.	353.8	382	382	382	382
Bore and Stroke, in.	3⅞x5	3⅞x5⅛	3⅞x5⅛	3⅞x5⅛	3⅞x5⅛
Compression Ratio	5.25:1	5.81:1	5.81:1	5.81:1	5.81:1
Rear Axle Ratio	7.36:1	7.14:1	7.14:1	7.14:1	7.14:1
Overdrive Ratio	0.794	0.788:1	0.788:1	0.788:1	0.788:1
Tire Size	10x20				
Gross Load During Tests, lb	22,000	22,000	22,000	22,000	22,000
Total Miles	154,000	159,000	160,000	150,000	101,000
Miles Since Carbon Removed	24,000	37,000	51,000	30,000	45,000



■ Fig. 25 – Influence of partially restricted cooling system on road octane ratings – truck No. 11197

PACKAGING and HANDLING

SINCE its first full-scale common-carrier air cargo experiment with a war-surplus four-engined 28,000-lb plane in 1919, the Railway Express company has done its utmost to advance cargo carrying by air. In 1927, it eagerly joined with the first group of transcontinental air mail lines simultaneously to offer transcontinental air express. The data gathered by Railway Express Agency during the ensuing 15 years it has worked with the domestic airlines provide one of the reliable sources of information. Some of the data given herein from this and other sources have been previously given to the industry, but is repeated here as a convenience for ready usage.

This paper relates to commercial traffic as differentiated from the cargoes being handled in connection with the war effort. It does not deal with what degree of prominency may be taken by contract-air-carriers or gypsy-air-truckers in post-war transportation. More specifically, it is related to scheduled operations within the United States, conducted by land airplanes flown by regulated common carriers.

Although this paper contains some data showing that many of the largest tonnages of traffic cannot be profitably flown, even with highly efficient planes in the post-war period, yet I am firm in the belief that air cargo within the United States has a tremendous future. Some 33 tons per day are being flown an average of about 1075 miles, producing a substantial revenue to the airlines. The comments and suggestions contained herein are the writer's personal opinions and should not be construed as reflecting the policy of his employers.

■ What Will Air Cargo Consist of

It is believed that almost all post-war air cargo domestic shipments will be of the same general nature as those now moving by air or surface transportation. While the speed of air transportation may eventually develop some new industries that will ship or receive their products extensively by air, there have been no such industries developed in the past and none are apparent in the immediate future. A near approach to such new business has been the special light-weight editions of periodicals for foreign subscribers that have been published here and in Europe, but these are in fact only a lighter and re-edited model of the publishers' standard products.

The vast amount of traffic that is moving in the United States by surface means is indicated in tables prepared and copyrighted by H. E. Hale of New York. Shown here are some of his figures for revenue in the United States, as well as some estimates relative to the rail traffic of Railway Express Agency and to parcel post:

	Year	Billions of Ton-Miles
Steam Railroads	1941	477
Great Lakes	1940	95
Pipe Lines	1941	60
Trucks	1941	44
Inland Waterways	1940	21
Railway Express Rail	1940	1¾
Railway Express Rail	1941	2¼
Parcel Post – Fiscal year	1941	4/5

Notwithstanding the statements made in the hope of headlines by publicity seekers, who have found in air cargo a new sounding board, the nature of most of this vast tonnage is such there is little probability that much of it will ever be carried by domestic airlines within the United States. The ton-miles carried by rail in 1941 were an all-time high – the greatest volume of traffic on record. Association of American Railroads data show the Class I railroad freight tonnage in 1940 was:

[This paper was presented at the SAE Air Cargo Engineering Meeting, Chicago, Ill., Dec. 8, 1942.]

THE relation of air cargo costs to volume of tonnage carried and the vital importance of keeping loading and unloading time down to a minimum are thrown into strong relief by Mr. Peterson in his summary of the experience of the Railway Express Agency in handling air express, and by his comments on classes of commodities and merchandise moved by air.

Rate of increase in air express, Mr. Peterson declares, is keeping pace with the rate of increase in passenger miles.

Special commodity rates, canceled for the duration, have helped develop tonnage, Mr. Peterson states, citing machinery, tools, and parts as the top group, representing about one-third of the total air express in one particular month.

Off-airline business will increase the use of pick-up equipment, and great possibilities lie ahead in utilizing the more than 300,000,000 ton-miles of unused capacity of passenger plane routes, which figure is more than 60 times the air express and freight flown in 1941. Spreading direct flying costs over the combined passenger and freight plane capacity in ships designed for the task is fundamental in lowering ton-mile costs for the post-war period.

Discussing handling costs, the author explains how the use of magnesium roller conveyers in 200 of the company's terminals is saving loading time, and he describes how three elevating means can

NG of AIR CARGO

by C. G. PETERSON

Chief Engineer, Railway Express Agency, Inc.

	%		%
Products of Agriculture	9.0	Products of Forests	6.1
Animals and Products	1.8	Manufactured Goods and Miscellaneous	27.7
Products of Mines	53.9	L.c.l. Freight	1.5

Of the above, bituminous coal ranked first in carload tonnage and revenue. In 1940, one-sixth of the freight revenues of the American railroads was produced by bituminous coal. In 1941, the average revenue per ton-mile of freight carried on the railroads was 9.36 mills, or less than 1¢ per ton-mile. The average revenues per ton-mile on railroads of the United States and Canada are the lowest in the world – with the single exception of Japan – as shown by the figures for the last years available, which for Great Britain were 2.4¢; Germany (State operated), 2.3¢; France, 1.95¢; Norway (State operated), 2.6¢. Practically all full-carload rail shipments are loaded and unloaded by the patron and at the patron's expense. It is impossible to think of air carriers setting out planes for shippers to load with their untrained employees, or leaving planes standing for unloading when it suits the consignee's convenience. In 1940, the average haul of all revenue freight on all railroads was 351 miles. Regarding the "load factor" in rail freight movement, for each loaded freight car moved 100 miles in 1941, the railroads hauled an empty freight car 64 miles, which indicates the "load factor" could not be more than 61%, but actually was appreciably less as many cars were not loaded to weight capacity. In 1940, the average load per car for all commodities was 37.6 tons; but of a breakdown of 17 commodities, only three were above this – these being bituminous coal, wheat, and corn, and the other 14 were below the average loading. In 1940, the railroads carried over 960,000 carloads of fruits and vegetables, a large part of the movement being under refrigeration. A considerable part of the perishable movement was heated to prevent freezing. Scanning the larger commodity breakdowns of rail freight traffic, it is apparent that outside of l.c.l. little of the rail freight can be expected to move by air.

be used for handling: portable elevators, trucks with elevated bodies, and inclined conveyers of the continuous-flow type.

THE AUTHOR: C. G. PETERSON has been chief engineer of the Railway Express Agency since 1938 and before that he was adviser in the Air Express Division of the company. From 1917 to 1920 Mr. Peterson was a lieutenant commander, USNR, engaged in engineering duties and postwar sale of surplus Navy equipment. Mr. Peterson has held positions with Wright Aeronautical Corp., Wright Flying Co., Aerial Advertising Corp., and Ford Motor Co. He was vice-president and general manager of the Martz Airline Co., when he developed and operated the direct New York-Buffalo route before selling it to American Airlines. Mr. Peterson has contributed articles on commercial phases of aviation to journals and magazines.

L.c.l. freight shipments are shipments too small to move on a carload rate. Such shipments usually consist of crates, cartons, boxes, barrels, furniture, castings, drums, sacks, and other types of miscellaneous shipments. Numerous class rates are used, some being higher than the first-class rate and many lower. In some instances the rates include pickup and delivery. At over 1000 cities, Railway Express has been selected by the local railroad as the most efficient means to perform this vehicle service or transfer between stations, just as most of the airlines have selected Railway Express. In all cases l.c.l. freight is handled into and out of the cars by railroad employees, and the carrier is responsible for damage occurring if mishandled. In a recent report prepared by a mid-continent rail carrier, it was worked out that for several typical movements, the first-class l.c.l. rate varied from 3.9¢ per ton-mile on a 907-mile haul to 6.6¢ on a 527-mile haul and 5.3¢ on a 2259-mile haul. A minimum l.c.l. charge is made which is almost invariably the charge for 100 lb even though the shipment weighs less. Estimates of the average weight per shipment vary from 250 to 350 lb.

Of the traffic on the Great Lakes, a considerable portion of the tonnage is coal and ore, frequently loaded into the vessels by car unloaders, which raise and tip over an entire carload. The unloading is accomplished by highly mechanized equipment. Grain boats are loaded and unloaded at completely mechanized storage elevators. Package traffic has diminished on the Great Lakes. In normal times, many automobiles are transported from the assembly plants, frequently as deck loads. Practically none of this traffic would ever move by air. The same conclusion is reached for the traffic moving by inland waterways and by pipe lines.

Considering the 44 billion ton-miles carried by trucks, it is only necessary to call on one's own memory of the character of traffic one sees in these trucks to realize what a small proportion would ever move by air. The trucking industry laid major emphasis on the advantage of door-to-door pickup and delivery. This was of added importance in short-haul traffic as the cost thereof was a larger percentage of the cost of the line haul.

Regarding the traffic of Railway Express, tests made indicate that of the 1,600,000,000 ton-miles of l.c.l. rail express moved in 1940, which comprise all classes of rail express, the first-class matter comprised less than half of the total weight – to be exact, 49.4%. The average first-class l.c.l. shipment weighed approximately 29 lb, against an average for all classes of 41 lb. As of October, 1939, 73.1% of all first-class rail express shipments carried charges between 25¢ and 99¢. For the first-class rail express shipments bearing charges of $1.00 and over, the average weight was 68.5 lb and the average charge, $2.69.

As of May, 1939, the average length of haul was 487.5 miles for l.c.l. rail express, and of this, the first-class rail express business that moved less than 350 to 450 miles produced 52.4% of the shipments, 52.9% of the weight, and 32.98% of the gross revenue.

Ton-miles are not a well-adapted yardstick for measuring express traffic. Consequently, the Express Agency does not compile data on that basis. However, it is estimated that in 1940 the rail express business had an average gross revenue or charge to the public of 10½¢ per ton-mile for all classes of l.c.l. traffic. Making suitable allowance for the rate increase authorized in January, 1942, would raise this figure to approximately 11¢ per ton-mile. For full carloads moving at that period by rail express, the gross transportation revenue is estimated at 3½¢ per ton-mile.

■ What is Moving by Air Express

The volume of domestic air express in 1941 was 5,242,529 ton-miles, and for the first six months of 1942 was 4,738,985 ton-miles – an increase of 120% over the similar six-month period of 1941. More ton-miles of air express and freight were flown in the pre-war years of 1937 and 1938 in the United States than were flown in any European or Central or South American country or Canada during those pre-war years with the possible exception of Russia. Furthermore, these figures on air cargo in the United States do not include excess baggage of air passengers, but such baggage is almost invariably included in European figures on cargo. Nor do the United States figures cited include the cargo flown to and from the United States on Pan American Airways, with which Railway Express Agency also has a contract.

The rate of increase in the ton-miles of air freight and express flown in the United States since 1935 until the end of the fiscal year 1941 practically equaled the rate of increase in revenue passenger miles for that period, and the rate of increase for the revenue to the air companies from air cargo exceeded the rate of increase of passenger revenue for that period. The rate of increase of air cargo during those years far exceeded the rate of increase of air mail both in ton-miles and in revenue to the air companies.

A number of analyses have been made by Railway Express in connection with increasing the traffic moved by air express. A very complete analysis was made in April, 1939, which included, in addition to the number of shipments of each commodity, the average weight and charge. An analysis was made in April, 1941, of the number of shipments handled in the various commodities. In Table 1, the percentage of shipments was actual for the month of April, 1941, but the average weight and average charge, and the percentage the weights and charges bore to the whole were computed from the unit data obtained in April, 1939.

The basis for rates for air express is controlled by the airlines with which Railway Express has contracts. The present air express rates set in 1934 are based on 4¢ per lb per 100 miles (80¢ per ton-mile), plus an arbitrary (fixed charge), which declines with both weight and distance. No arbitrary is in the rates for weights over 25 lb or for distances over 2349 miles, above which distance there is no increase for additional mileage. Valuation charges are 10¢ per $100, the same as in rail express. Higher charges are made on shipments with specific gravity less than 4 1/3 lb per cu ft. Certain commodity rates at appreciable discounts were set (all commodity rates are now canceled), and

Table 1 – Air Express Analysis – April, 1941
Commodity Groupings

Commodity Group	Average Weight	Average Charge	% of Total: Shipments (Actual)	% of Total: Weight (Computed)	% of Total: Charges (Computed)
Machinery and Hardware	10.0	$4.33	23.26	31.67	32.77
Printed Matter	13.6	3.36	15.11	28.08	16.62
Store Merchandise	5.0	2.78	13.39	9.25	12.08
Motion Picture Films	9.5	5.28	4.32	5.62	7.42
Electrotypes and Matrices	4.2	2.04	6.11	3.51	4.07
Cut Flowers	5.0	2.91	3.63	2.48	3.44
Valuables	3.8	1.84	8.25	4.24	4.95
Miscellaneous	6.0	2.99	2.44	2.01	2.37
News Photos	1.3	1.21	4.42	0.79	1.73
Drugs	5.4	2.53	1.46	1.08	1.20
Transcription Records and Radio Parts	3.3	1.90	4.53	2.04	2.00
Freight Manifests	2.7	1.52	4.36	1.61	2.15
Jewelry	2.1	1.64	2.60	0.75	1.30
Food and Raw Samples	4.2	2.35	1.78	1.02	1.36
Optical Goods and Cameras	4.5	2.39	1.85	1.14	1.44
Personal Baggage	15.1	5.49	2.19	4.52	3.91
Liquor	5.1	2.95	0.30	0.21	0.29
Total	7.3	$3.07	100.00	100.00	100.00

charges for planeloads when the patron provides vehicle service to and from the airport are set by individual airlines at various levels. Thus there is no set figure per ton-mile for present l.c.l. air express or planeload air freight. In 1941, the gross air express revenue was at the rate of 81½¢ per ton-mile for the entire system and on all classes of traffic. The net payments to the airline companies in that year were at the rate of 55.4¢ per ton-mile. The amount retained by the Express company for its services and the use of its facilities was 9.62% of the gross revenue. The amount spent for advertising, insurance, damages, vehicles, and handling was 22.7% of the gross revenue. The average weight per shipment was 8.6 lb and the average charge was $3.28. There were 1,306,629 air express shipments and a gross revenue of $4,277,070, which was an increase of 40.5% over 1940. Net payments to the air companies were $2,894,000. The weight of the shipments was 5620 tons, and it is estimated the average length of haul was approximately 935 miles. Of the total shipments, it is estimated 256,755 were from off-airline points, and the air revenue on these shipments was $1,036,407. From this it is derived that the shipments which moved between airport cities had an average revenue of $3.09, while those that moved to or from off-airline cities had the much higher average air revenue of $4.04. The ratio of off-airline shipments to shipments between airport cities was as 1:4.1. The ratio of gross air revenue on off-airline shipments to air revenue on shipments between airport cities was as 1:3.1.

The most recent figures for 1942 on air express show for August an average weight of 17.6 lb, and preliminary estimates for September, 129,026 shipments – an increase of 8.8% over September, 1941, with a gross revenue of $923,028 – an increase of 129%, which gives an estimated average revenue per shipment of $7.15. In this month, the ratio of off-airline shipments to shipments between airport cities was as 1:3.5, and the ratio of air revenue on these off-airline shipments to the revenue on shipments between airport cities was as 1:2.7. The average air revenue on these off-airline shipments was $8.64.

Commodity air express rates at appreciable reductions from the base rate, as established by the airlines, of 4¢ per lb per 100 miles, which is 80¢ per ton-mile, have been filed with the Civil Aeronautics Board by the Express Agency from time to time. One of these rates produced a good volume of valuable traffic. Another produced no traffic at

all. All such commodity rates were canceled during the summer of 1942 and may not be re-established until after the war. The most widely used commodity rate was the 40% discount given on newspapers and magazines. This represents a base rate of 48¢ per ton-mile. Exact reports on the quantity of newspapers and magazines shipped at these reduced rates are not maintained, but a complete tally made for one month indicated for that month 10% of the total tonnage moved under this commodity rate and produced 2.65% of the gross revenue. Several commodity rates on cut flowers were put in effect from West Coast points at approximately 60% of the base rate or 48¢ per ton-mile. Apparently during the spring of 1941 this traffic averaged about 4500 lb per month for $1850 revenue, but this may not have been all new business.

The rate on lobsters from Boston of approximately 50% of the normal rate was established in the spring of 1942; and, even though this was at the rate of about 40¢ per ton-mile, no traffic was obtained.

The lowest commodity rate established was on seafood from North Pacific points at about 26½¢ per ton-mile. After several months of intensive development and advertising, this produced about 625 lb per month of new business for an average monthly revenue of about $130.

■ Where Air Cargo Moves

Experience has shown that the heaviest flow by volume by air express is from East to West and from North to South. Even at the present time this direction of flow holds good. Table 2 shows examples of this direction of flow among five states.

Table 2 – Direction of Flow of Air Express for Typical States by Weight – April, 1939

Forwarded from	Total Forwarded to All States lb	To Destination: New York lb	California lb	Illinois lb	Florida lb	Texas lb
New York State	102,082	1,967	15,240	18,970	11,112	4,218
California	74,409	11,309	32,421	2,194	595	2,172
Illinois	44,012	13,025	4,043	186	1,503	2,764
Florida	6,443	3,212	311	245	361	112
Texas	7,104	1,033	396	495	196	1,415
Total Received From All States	397,719	60,037	68,835	36,542	20,950	19,269

It has been found that the movement of traffic between cities shows possibly greater fluctuations of flow than between states. In response to frequent inquiries regarding the flow of traffic in rail express business, Table 3 has been prepared from previous tests made; and, for your convenience, there is shown below the tonnage figures, in parentheses, the pounds of air express that moved between these same cities in April, 1939. It should be remembered in examining these figures that these 14 cities do not represent the 14 cities with the heaviest air express traffic. Miami is one such city. While the rail express business moving at present is greater than in May, 1939, yet the percentage of increase is not as great as that for air express. For August, 1942, the total weight of air express of 2,048,085 lb was 516% of the 397,719 lb flown in April, 1939. In considering the poundage of rail express shown on the accompanying table, it should be noted that it covers shipments originating at or destined for the individual cities mentioned as shown on the waybill. These figures do not include transfers made at those cities, or, in general, suburban traffic for the cities. The usual limits are the corporate boundaries of the cities concerned. Similar data are not available for l.c.l. freight shipments moving by rail, but it is known that such shipments do not follow the pattern of rail express shipments. For instance, at New York the volume of inbound l.c.l. freight shipments far exceeds the volume of l.c.l. outbound shipments.

Table 4 has been worked up to give some idea of the relative distance flow of express traffic, both in the rail service and in the air service. Data are not available to use the same measurements of volume of the traffic for both services. It will be noted that for the rail traffic the data are given in percentage of the total weight by the distance increments. For the air traffic the weight was not readily obtainable, but the percentages of shipments and revenue given indicate the volume flown for the various mileages. For both of these services, the high quantities handled in the lower mileages will be noted. Undoubtedly much of the short-haul business that was flown was because the consignee desired same-day or same-night delivery. The frequency of service on the airlines contributed largely to building up this short-haul volume.

Table 3 – Tons of Rail Express Traffic for May, 1939, between 14 Cities with Greatest Population (In Parentheses, Pounds of Air Express Traffic for April, 1939)

To \ From	Baltimore	Boston	Buffalo	Chicago	Cleveland	Detroit	Los Angeles	Milwaukee	New York	Philadelphia	Pittsburgh	St. Louis	San Francisco	Washington
Baltimore	....	123.8 (10)	5.3 (0)	62.1 (673)	13.2 (39)	15.5 (244)	3.1 (251)	6.1 (34)	385.3 (423)	81.4 (9)	10.1 (43)	10.5 (17)	2.0 (61)	25.8 (30)
Boston	36.7 (43)	....	22.4 (46)	117.8 (592)	29.6 (313)	27.3 (250)	8.0 (145)	11.6 (80)	629.8 (1579)	95.8 (216)	13.1 (56)	14.8 (76)	8.8 (130)	15.9 (50)
Buffalo	20.3 (21)	95.0 (127)	....	66.5 (257)	37.9 (28)	24.4 (436)	2.6 (104)	8.0 (41)	270.1 (669)	28.1 (75)	10.4 (48)	13.7 (12)	9 (2)	2.9 (5)
Chicago	64.8 (277)	142.8 (46)	36.7 (206)	....	105.1 (717)	105.5 (1373)	31.1 (1059)	77.2 (0)	1375.8 (17,635)	173.7 (1012)	29.5 (258)	81.5 (115)	28.9 (750)	24.8 (202)
Cleveland	19.5 (71)	70.2 (690)	45.5 (17)	205.2 (698)	....	47.5 (750)	5.7 (149)	12.9 (132)	427.0 (2257)	62.5 (247)	33.3 (374)	30.6 (2)	4.5 (40)	8.0 (131)
Detroit	20.8 (46)	67.9 (671)	20.0 (537)	294.4 (1847)	69.2 (1992)	....	9.5 (733)	16.8 (583)	484.6 (7817)	59.8 (683)	14.2 (179)	40.2 (59)	4.9 (339)	6.0 (111)
Los Angeles	7.6 (79)	8.0 (119)	2.9 (22)	70.2 (1977)	12.5 (504)	14.7 (673)	....	6.1 (111)	277.5 (8612)	19.7 (991)	3.1 (58)	9.1 (404)	132.4 (2106)	2.2 (271)
Milwaukee	7.7 (9)	27.3 (51)	3.8 (0)	224.4 (9)	19.2 (72)	16.0 (535)	4.5 (138)	....	168.9 (1354)	17.4 (51)	4.2 (66)	19.1 (16)	2.1 (41)	2.0 (7)
New York	215.9 (81)	323.7 (821)	106.9 (677)	712.8 (10,271)	175.2 (3572)	143.7 (3685)	92.4 (8343)	77.4 (709)	....	1017.5 (1019)	116.4 (1183)	119.5 (676)	80.4 (1761)	141.4 (1489)
Philadelphia	99.5 (5)	124.8 (66)	16.0 (149)	168.1 (1447)	35.4 (267)	36.3 (345)	6.4 (1395)	13.5 (118)	1342.1 (1257)	....	25.5 (808)	26.6 (69)	3.3 (70)	24.7 (2)
Pittsburgh	36.4 (24)	41.3 (41)	12.7 (84)	124.4 (443)	54.0 (380)	27.5 (271)	3.6 (106)	11.6 (61)	389.1 (982)	97.0 (67)	....	16.2 (20)	1.5 (98)	16.1 (203)
St. Louis	14.2 (60)	40.9 (161)	5.8 (38)	179.3 (514)	27.3 (124)	24.9 (223)	7.7 (176)	13.5 (8)	345.7 (2512)	49.1 (104)	6.5 (65)	....	5.4 (320)	17.9 (18)
San Francisco	7.5 (72)	8.7 (107)	2.3 (184)	53.4 (952)	7.5 (279)	8.1 (196)	95.9 (21,925)	6.5 (76)	205.5 (4525)	13.4 (439)	1.5 (67)	6.5 (89)	....	7.1 (125)
Washington	81.5 (0)	140.6 (81)	9.3 (46)	76.5 (735)	17.4 (166)	17.0 (363)	6.5 (267)	8.3 (26)	473.8 (2810)	105.5 (109)	12.7 (421)	15.4 (36)	8.2 (186)	....

Table 4 – Distances of Express Movements for both Rail Express and Air Express

Miles	L.c.l. Rail Express, % of Total Weight as of May, 1939	Air Express, % of Total as of April, 1939	
		Shipments	Revenue
0 to 349	56.6	15.8	8.77
350 to 649	18.0	18.1	12.50
650 to 949	13.1	24.2	20.14
950 to 1249	5.9	12.6	12.61
1250 to 1549	2.9	5.4	6.09
1550 to 1849	1.2	3.4	4.36
1850 to 2149	0.6	6.1	8.56
2150 and over	1.7	14.4	26.97
	100.0	100.0	100.0

■ When Air Express Moves

Frequency of airplane service has been one of the factors that has led to the tremendous percentage increase of air express traffic. Table 5 indicates the frequency of service prior to the curtailment due to war conditions.

Table 5 – Air Express Schedules Each Weekday Starting from New York for Other Important Air Cargo Terminals As of Jan. 5, 1942

New York to		New York to	
Chicago	30	Dallas	11
Los Angeles	20	Washington	37
San Francisco	14	Boston	21
Miami	5	Seattle	22
Detroit	16	Milwaukee	24
Cleveland	16	Twin Cities	24
St. Louis	8	Kansas City	12
New Orleans	8		

From a survey made in March, 1941, at 17 of the larger airports of the country and for all airlines with the exception of one principal north-and-south line, the number of flights on which air express could be flown is as given in Table 6, together with the percentage of the weight of air express that was on board the planes at that time.

Table 6 – Number of Airplane Departures from 17 Airports During March, 1941, by Time Intervals of Departure and Percentage of Weight of Air Express on Departing Planes to Total Carried

	Average Number Airplane Departures	% of Weight of Air Express
Midnight to 4 a.m.	30	11.7
4 a.m. to 8 a.m.	56	14.3
8 a.m. to noon	99	10.0
Noon to 4 p.m.	132	16.4
4 p.m. to 8 p.m.	133	19.4
8 p.m. to midnight	78	28.1

Tuesday, Wednesday, and Thursday are usually the heaviest days of the week for handling air express, as shown by tests made in the pre-war period at La Guardia Field. From 1934 to 1940, the highest poundage month fell three times in October, twice in September, once in November, and once in December. The highest poundage month for these years averaged a total tonnage of 36% more than the average tonnage per month for the year.

■ Who Ships by Air

According to a traffic expert well-versed in both air and surface transportation, the pre-war air express traffic consisted largely of articles or goods used in the process of production rather than of ordinary consumer goods. The speed of the airplane is often of particular value to the former; but, except in certain circumstances, of little specific value to the latter. This is due to the fact that consumer goods are generally produced in mass lots, and there usually is a period of storage either preceding or following distribution, or both; also a period of display before sale to the retailer, and, in most instances, a period of storage and display by the retailer before sale to the consumer. This generality is well borne out by the commodity grouping in Table 1, from which it will be seen that in the pre-war days:

Machinery, hardware, and the heavy industries rated the highest in the actual number of shipments and percentage of total weight and charges. These shipments were by no means all repair or emergency parts, but, instead, the bulk of them probably were shortages required in production. The regularity with which these heavy industries furnished the most profitable shipments for air express forecasts that in the post-war period they will make up an equally important factor. These industries under good business conditions work 24 hr, many of them seven days a week. They ship both day and night and demand immediate deliveries.

Printed matter, the next most frequent, heaviest, and most profitable commodity, includes newspapers and periodicals. The heaviest individual series of shipments ever made in the pre-war period was of magazines for nationwide distribution. Newspapers have moved between various cities in large volumes, and this traffic can be expected to increase. Magazines and newspapers can be considered as consumer goods and the shipment directed by the shipper. They move at various times of the day. They can be dropped by parachute, and frequency of schedules is particularly important in their movement. Many other items in the graphic arts industry are moved against time to fit into predetermined production schedules of sales, advertising, and public events in that fast-moving industry.

Electrotypes, matrices, and plates come under the same general classification as printed matter, but they are shipped not as consumer goods but as a cog in the wheel of the printing industry. These move nationwide both day and night, and immediate delivery is required.

Store merchandise has probably received more intensive sales effort than any other commodity moving by air. The results have been that while these shipments only accounted for 3 1/3% of the shipments in 1934, they grew to 13 1/3% in 1941. An appreciable volume of the l.c.l. freight and the first-class rail express business consists of this store merchandise. On the extent to which a greater volume of this will be flown may depend the degree of the success of post-war air cargo. Even though most retail establishments are set up on the basis of buying in sufficiently large lots to secure wholesale prices and maintain a complete stock, yet in many of the high-pricéd specialty women's wear stores at considerable distance from the manufacturing centers, the practice of carrying skeleton stocks, making sales with model gowns displayed by mannequins, and when an order is placed by a customer, obtaining the desired color and size by air for next-day delivery direct to the customer has become increasingly frequent and proved profitable for the stores. There are limitations to this, of course. On the other hand, smart merchandising may lead to an expansion of this principle, because most stores make delivery of purchases the following day; and as air express can reach practically the entire country equally fast, there is every hope that this business may continue to increase. Buyers filling in shortages on fast-moving lines add to the volume of store merchandise moving by air. These shippers apparently cannot be induced to use other than evening departures. They are content with daytime deliveries.

Motion picture films moving regularly by air consist

largely of the newsreels which are shipped principally from New York twice a week. These require very exact handling to reach the theaters at not later than the appointed time, whether holidays or nights. Frequency of schedules and dependability of arrival are of paramount importance in this traffic. The traffic managers are well-versed in the airline schedules, and as the prints are completed and packed, they are dispatched in sequence according to the distance to be traveled. Practically all the newsreel companies ship on the same days and during the same hours. The quantities fluctuate depending on the news value of the releases.

Cut flowers have always been a good source of air traffic, and in the post-war period the volume of this business should increase to large proportions. The bulk of this traffic is from growers and wholesalers to individual shops or to the wholesalers in the destination cities. This business is especially important in that it moves from California to the East, from Texas to the North, and from Florida to along the Atlantic seaboard and the Middle West. It constitutes one of the exceptions in the normal flow of air or rail express traffic. Growers are willing to cooperate on time of shipments, but there are limitations due to daylight picking. Most of the shipments require scheduled deliveries which cannot be met unless there is continuity of flight. Individual shipments of flowers as gifts or for funerals require exact timing, and delays cause ill will as well as the loss of potential patrons. More loss and damage claims and claims for delay arise from cut flowers than from any other commodity.

Valuables, jewelry, optical and photographic goods have always been an important part of the express business, and in air express they constitute an appreciable volume. Their handling requires protection, and serious losses have occurred. Much of this traffic is closely timed and either for protection or for time of delivery requires special handling.

News photos and drugs do not comprise as large a percentage of the total of air cargo as they did in the early years. However, the volume of each has grown appreciably, and they contribute a worth-while percentage of the gross revenue. They are invariably timed shipments, and frequency of schedules is important in their handling.

Transcription records is an item that showed a very rapid growth in the pre-war period. Shipment by air is well suited to the handling of this scheduled merchandise. Unfortunately, they are exceedingly fragile and require careful handling. They are invariably a timed shipment, and frequency of schedules has done much to build up this traffic.

Personal baggage has shown a high average weight and revenue per shipment. It is one of the few commodities that actually are shipped by air by the man-on-the-street or the lady-of-the-house, and can therefore be classed as consumer shipments. However, many of these shipments of personal baggage have been in connection with other forms of transportation – to meet steamship departures, or lost baggage and laundry, or articles left at hotels or in transit.

Very little food has moved by air for actual sales purposes. The majority of such shipments have been for publicity purposes, or as samples or gifts. However, samples of food, together with raw samples of wool, cotton, nuts, coffee, silk, oil, and other commodities have formed and will continue to be a very important part of air cargo. These are not timed shipments. Frequently many are shipped at one time. They move counter to the flow of manufactured goods.

■ Why Shipments are Made by Air

As will be seen from the foregoing tables, practically all air shipments in the pre-war period were commercial shipments. The same may be expected in the post-war period. The only reason that such shipments are flown is that the speed of flight returns a profit to the patron who pays the charges. In the majority of cases it has been found that it is the consignee who specifies whether a shipment is to be flown or sent by other means. With the exception of some of the cut-flower business, the newspaper and magazine traffic, and a few other scattered instances, it is the consignee, not the shipper, who makes the decision as to whether the goods are to go by air or otherwise. The necessity of selling the consignee constitutes an important difference in the methods required to build up air cargo business and the sales methods and activities used for obtaining passengers or air mail. The consignees of the country are located in the million and a half retail establishments throughout the length and breadth of the land. Selling these consignees requires not only personal acquaintance and some knowledge of their individual businesses, but also frequency and persistency in sales effort. As previously stated, many of the bigger and better shipments move to or from points that are not located directly on the routes of the passenger airlines. This off-airline business shows the greatest rate of increase. With the perfection of equipment for picking up shipments while the airplane is in flight and discharging them without the airplane landing – as is now being done among the existing feeder-line routes – this off-airline business may be expected to increase. But selling the consignees in the smaller towns on feeder-line routes is a hard field to harvest, as frequency of service is not as great as on the better-traveled air routes, and the high revenue shipments are of a size, weight, or commodity that cannot always be safely handled without landing.

In selling the million and a half potential consignees of air cargo, one of the points of greatest sales resistance has been the degree of dependability of the shipments arriving at the time desired. The coordination of rail express service with the air express service, as conducted by Railway Express Agency, has been an important factor in breaking down this item of sales resistance. The consignee knows that the shipment will continue to move by the fastest available means and will not be subjected to an avoidable delay. At some times of the year, when flying conditions are adverse, this coordination of the traffic between rail and air express has comprised a very large percentage of the air shipments. Even when flying cargo planes with no passengers at all, it cannot be expected the airline executives will permit undue risks to be taken with their highly valuable airplanes, their personnel, and their cargo. Therefore, it appears essential for the growth of air cargo that the consignees can be truthfully assured their goods will continue to move toward them by the fastest means available regardless of interruptions to the air service.

■ Future of Air Cargo in Passenger Airplanes

The CAB supplemental answer to Senator George, made public in September, 1942, included some data as to the possible growth of air transportation in the succeeding

four years if the war had not interfered. In general, this was to the effect that six billion revenue passenger miles on domestic airlines in 1946 would appear to be a conservative forecast. If an average annual operation of 400,000 miles is assumed per aircraft, and a load factor averaging 60%, this would mean aircraft capacity of some 25,000 seats. This is 10 billion seat miles per year which, on the basis of 10 passengers and their baggage to a ton, represents one billion ton-miles capacity. The 60% passenger load factor assumed leaves 40% unused capacity or 400 million ton-miles unused capacity available for cargo.

If all first-class long-distance mail were sent by air plus the present volume of air mail, the total has been estimated by some as between 70 and 100 million ton-miles per year. Deducting this mail traffic would leave between 300 and 330 million ton-miles of unused capacity available for air cargo. The volume of air express in 1941 was 5¼ million ton-miles. Thus there would be unused weight capacity available for air express and freight in addition to capacity for surcharged and unsurcharged mail equal to an increase of some 6000% of the ton-miles of air express and freight flown in 1941.

Similar conclusions are arrived at through a study of the survey of the unused capacity available in airline passenger planes, made at 17 of the larger cities in March, 1941, by all of the airlines except one prominent north-and-south line. Such a study reveals that at that time the volume of air express and freight could have been increased approximately 22 times and still be within the permitted gross weight for the average departure.

An examination of the individual flights and route data in the Station-to-Station Airline Traffic Survey of the CAB for September, 1940, goes far to bear out the principle outlined above – that on most flights an appreciable capacity is available for handling air cargo in addition to the average passenger load. For the fiscal year 1941, the revenue passenger load factor was 56.5%. The average pay passenger load was 9.68. Table 7 roughly indicates the average loading.

Table 7 – Average Load per Revenue Airplane Mile Fiscal Year, 1941

	Lb
Pay passengers: 9.68 per plane	1936.0
Mail load	293.3
Express load	68.2
Excess baggage load	18.4
Total Average Load	2315.9

If one assumes the average pay load of a 21-passenger airliner at 4500 lb, it is seen that there was an unused capacity of 2185 lb, which is 32 times the average amount of air express actually carried.

It is a well-known principle that there is practically no difference between the direct flying cost of a plane partly loaded and a plane fully loaded. Hence the airlines that are in a position to carry air freight and air express on their regular passenger and mail planes have an opportunity to carry a vast volume of such traffic with little or no additional direct flying cost. This fact is so fundamental that, in the opinion of many, it will be one of the major factors in determining the amount and character of air cargo that will move in the post-war period.

To assure the type of equipment that can handle this volume of air cargo in the passenger planes economically and without detriment to schedules or causing inconvenience to passengers, is one of the fundamental points which must be considered by aeronautical engineers and manufacturers in designing airline planes for that period. The packaging for, and handling of, air cargo in and out of these planes must be included in such a study. To my mind it is reasonable to expect that at way-points between terminals the packaging and handling should be such that the detention time for the air cargo will not exceed the detention time for fueling, provisioning, and passenger interchange of, say, 10 min per stop. Any detention time at way-points longer than this must be charged against the air cargo operation, but the writer is convinced that since the design of post-war passenger planes is still in flux, and the shape of the fuselage, size and number of doorways, size and accessibility of passenger and cargo compartments, and convertibility of space between passengers and cargo are none of them limited by the fixed factors which regulate the size, length, weight, and height of over-the-road trucks, the gage and height of railroad cars, or the hatch and multiple-deck construction for ocean-going vessels, that the aviation industry, if it gives due attention to this problem, can develop vehicles that will be able to earn a substantial revenue through the otherwise unused capacity of passenger planes.

■ Packaging of Air Cargo

More work has been done on the packaging of cut flowers for transit by airplane than on any other commodity. Extensive tests have been conducted on the effects of freezing, high altitude, and high temperature. Table 8, based on data supplied by the Department of Agriculture, indicates that flowers are not seriously damaged at 32 F.

Table 8 – Freezing Point of Flowers

	Petals, F.	Leaves, F.
Anemone	28.1	....
Carnation	28.4	27.4
Chrysanthemum	28.4	29.6
Easter Lily	27.5	29.2
Gardenia	28.3	....
Gladiolus	28.7	26.8
Orchid	30.8	....
Peony	29.0	28.4
Ranunculus	28.6	....
Hybrid Tea Rose	30.0	28.3

Manufacturers have offered a variety of methods of dry packing to protect cut flowers from freezing during transportation by air or by surface means. However, since the specific heat of cut flowers is low, insulation to retain the original heat is not sufficient; and, as flowers are rendered practically worthless if exposed even for a short period to temperatures much below the freezing point, the insulation methods of packing have not afforded dependable protection. These dry packings included dry cotton batting, balsam wool, dry newspapers, aluminum foil, shredded newspaper, and moss.

In the fall of 1936, Railway Express initiated a series of experiments on a "wet pack" method, utilizing the latent heat of fusion of water to provide close to the blooms a gentle source of heat which would not become effective until the temperature within the package dropped to 32 F. Then, as the water in the "wet pack" froze, 144 Btu were released for each pound of water frozen. Enlisting the aid of the Department of Agriculture, that organization ran a series of cold-room tests with varying amounts of water in packages of flowers 30 x 10 x 6 in., or 1800 cu in., with an area of 1080 sq in., at 15 F room temperature. With 5½ lb of water absorbed in 15 thicknesses of news-

paper, it required 23½ hr of exposure to lower the temperature to 30 F, and 48 hr to reach 28 F. With 10½ lb of water absorbed in 30 thicknesses of newspaper it took 41 hr to reach 30 F, and 70 hr to reach 28 F. The same size packages in a cold room of 0 F, with 5 lb of water reached 30 F in 8 hr, and 28 F in 10 hr. With 10 lb of water, the temperature dropped to 30 F in 22 hr and to 28 F in 24 hr. This latter test was exceedingly severe for air cargo as while 0 F might be experienced in a plane for a few hours, normal shipments would practically never be subjected to such a low temperature for as much as 8 hr. These cold-room tests were so promising that early in 1937 a series of experimental flights with cut flowers protected by the "wet pack" method were made from the West Coast. The first group of these test shipments loaded in unheated compartments were flown to Washington accompanied by similar shipments packed in the conventional dry-pack methods. The lowest external temperature reached was 4 F. With the "wet pack," the flowers were unharmed, the lowest inside temperature being 32 F. The dry shipments were all frozen. During January and February of 1937, a dozen or so similar test shipments were made from San Francisco to New York, some via Seattle, these shipments being loaded in the unheated nose of D-247 planes. Record was kept of the external temperatures at various points for each trip, and uniformly the shipments protected by the "wet pack" came through with little or no damage from freezing. Cotton batting was found a better reservoir for water than newspaper, as cotton would absorb and hold about 11 times its weight in water, whereas newspaper would only absorb about 3½ times its weight. It was thought that a commodity tariff could be worked out so the shipper would not be required to pay the high air express rates for the weight of the water used as the heating element, but with the introduction of the DC-3 in which flowers could be carried in a heated compartment, this commodity rate was not promulgated.

One of the largest West Coast shippers now has his girl veteran packers pack gardenias without touching the petals by tucking the freshly picked blooms into waxed cardboard cartons with a packing of cotton around each flower. The flowers are then sprayed with an overhead nozzle, then another layer of cotton placed above them and the whole slipped into the shipping carton. Twenty-five or 50 blooms are the usual number per pack. Some growers staple the stem to a cardboard liner. Invariably the cotton is well dampened, but not approaching saturation.

Orchids are shipped with each stem placed in a test tube of water with a special cork that will not bruise the stem. Each is stapled to cardboard by paper-covered twisters, adhesive tape, or stitched with cord, and the whole protected with finely shredded paper between and around the petals. Many orchids are grown under glass in the vicinity of New York. When these are shipped by air, the average length of flight is not as long as for the flowers from California.

Early in 1937, one Florida shipper offered to deliver 1000 sweet peas for $15.00 including air express charges to airline points in the North and East. At that season space limitations in the northbound planes were such that these could not be carried in the heated compartments; and, as the competitive price was too close to permit the use of the "wet pack" method, the business could not be developed.

To ascertain if altitude had any ill effect on flowers, the Department of Agriculture ran a series of tests in a vacuum chamber with the pressure reduced to the point corresponding to about 20,000 ft altitude. The flower petals and leaves were unharmed by this low pressure. In the spring of 1941, United Air Lines made a special flight from Oakland to Reno in a DC-3 plane to test, among other characteristics, the effect that high-altitude flights would have on flower shipments. The Superintendent of Loss and Damage of the Express company participated in the test. The climb to 25,000 ft was followed by a rapid descent of 11,000 ft in 7 min. It was found the change in atmospheric pressure did not disturb the flowers from the position in which they were placed within the boxes. It did not cause the staples holding the flowers to pull loose. There was no bursting of the cells in the petals of the gardenias and orchids. It was however noted that when heat-sealed cellophane wrapping was used, the variation in air pressure caused an expansion and collapse of the cellophane envelopes. It was demonstrated this could be overcome by venting the cellophane envelopes.

Regarding protection of cut flowers from overheating, most flower merchants maintain 45 F in their cold-storage rooms. It is desirable not to exceed 50 F in carrying cut flowers. It is concluded that exposure of packages of cut flowers to temperatures ranging upward from 50 F, particularly above 60 F for an extended period of hours, would have an adverse effect on the marketable condition of the flowers. This is particularly true for varieties so packed that the weight per cu ft is greater than is customary with gardenias and orchids. Flower boxes should be kept right-side-up to prevent internal crushing or bruising of the blooms. With cargo airplanes, it should be possible to utilize water ice or dry ice for refrigeration in cases where the CO_2 gas would not harm the lading. In rail service, many flowers move East from California with a 10% allowance for water ice. The ice is wrapped in newspapers with an outer covering of waxed paper. This ice package is then placed midway between the ends of the shipping box. Some Los Angeles shippers include a small piece of dry ice with the water ice. Careless wrapping of the ice at times results in soaking and damaging the containers, with a corollary risk of wetting adjacent packages. A suitable method of confining water ice could be devised.

In rail express service, flowers from the West Coast are moved East under refrigeration during the warm months of the year. During the winter months they are handled in dry refrigerator cars heated by drums of hot fresh water which are changed each 12 or 24 hr along the route. Charts have been worked out to indicate the number of drums of hot water to be placed in each car according to the temperature forecast. Like the "wet pack" method, utilization is made of the 144 Btu per lb of the latent heat of fusion of the water in the barrels which equals the amount of heat given off by the water as it cools from 176 to 32 F.

The effect of a change in altitude on cans with various types of covers was tested on the high-altitude test flight made by United Air Lines. In all, 28 cans from a pint to a gallon in size were tested, some being ⅞ full of water and some ¾ full. All cans, particularly those that were ⅞ full, showed a pronounced bulge at a 5000-ft elevation. The bulge increased greatly with the rate of climb, and at 5000 ft the maximum expansion appeared to be reached. Screw top cans, pressure top, and single and double friction tops were tested. It would appear that with the exception of single friction top cans, all other types of closures may

be transported without undue risks when filled and properly closed. From 10,500 ft up, all single friction top cans showed evidence of leakage.

Radium, thorium, and other radioactive substances including salts and liquids containing radium moved in considerable volume in air express service until it was found they might fog and thus damage undeveloped and unexposed films including moving-picture and X-ray film. Considerable work was done in endeavoring to ascertain the thickness of a lead shield that would be required to afford protection to photographic films at different distances and for varying lengths of time in transit. A large shipper reported about 67½% of the radium shipments were 25 mg or less. The usual protection for this amount was 3 mm of platinum and 2 mm of lead. As the density of platinum is about twice that of lead, this would be equivalent to about 5/16 in. of lead. However, that did not give sufficient protection to films in the mail. Government tests indicated that 25 mg of radium at the center of a 1½-in. diameter sphere of lead would not fog films during 10 hr of exposure at 4 ft and 40 hr at 10 ft. Radioactive substances lose strength rapidly; hence speed of transportation is important. Furthermore, speed of transit permits wider use of the limited basic supplies. As radioactive materials are not now permitted to be flown in mail planes, this is a commodity that could be flown in two directions in exclusive cargo planes if kept away from films.

Fur-bearing animals including chinchillas are flown from time to time to and from the breeding ranches. Occasionally, dogs and other pets have been flown, but it is not convenient to handle and give proper care to live animals in combination passenger planes. Baby chicks and turkeys are handled in great volume in rail express, and these might constitute a commodity to be flown from California to the Middle West. They are vulnerable to drafts and excessive heat or cold. Pan American Airways has flown large quantities from Miami to Caribbean points that could be reached in less than 48 hr from the time of hatching.

■ Specific Gravity of Air Cargo

From a series of tests made in March, 1941, at six of the large airports of the country when 95,512 lb of shipments were weighed, it was found that the average weight per cu ft of air express at that time was 13.3 lb. The figures were not essentially the entire traffic handled at these airports during the test period. In detail, the specific gravity of various types of air express at the six airports in March, 1941, was as follows:

	Lb per cu ft
Cut Flowers	4.9
Millinery	2.7
Other Merchandise	14.1
General Average of All Commodities	13.3

In response to numerous inquiries relative to the present-day average weight of air express shipments, a spot test was made at La Guardia Field in November, 1942. For the purpose of this test a number of shipments were individually measured and weighed. It is believed that designers and operators will find these unit weights of various commodities useful in calculations for post-war traffic by combining them with the percentage of commodities expected to comprise the post-war lading. Since these shipments were measured individually, allowance should be made for voids when they are stowed and secured for transit in the airplanes.

Table 9 – Specific Gravity of Various Airborne Commodities, La Guardia Field, New York, November, 1942

	Lb per Cu Ft
Printed matter	35.6
Magazines	34.9
Bank shipments	25.7
Blueprints	18.4
Electrotype plates, matrices	20.9
Photographs (bulk)	34.6
Films in cartons	49.0
Radio Records – glass in carton	9.3
Yeast	111.5
Store Merchandise	9.0*
Shoes	9.0
Hats	1.5
Furs	4.6
Personal	15.9
Chemicals and Serums	11.4
Jewelry, watches, and lenses	15.1
Heavy industries – hardware and tools	36.9**
Aircraft materials	22.7***

* Store goods ranged from 4 lb per cu ft for lingerie to 21.3 lb for piece goods.
** Heavy industry products ranged from 8.4 lb for V belts and 8.3 lb for marine engine parts to 150 lb per cu ft for tools.
*** Aircraft materials ranged from 7 lb per cu ft for magnetic material to 62.4 lb for parts.

Under the present domestic air express tariffs, standard rates are applied when the cubic measurement of a shipment does not exceed 400 cu in. per lb. Over that, charges are made on the basis of 1 lb for each 400 cu in. Thus shipments with specific weight of less than 4.3 lb per cu ft are charged for by volume.

■ Handling of Air Cargo

It will be clear from the foregoing that there is every reason to expect a large portion of the post-war air cargo will be carried in passenger planes. Undoubtedly mail will also be carried in these planes. But the following suggestions will apply for either exclusive cargo planes or combination planes. To obtain a fast turn-around of the planes and provide for their being in flight a maximum number of hours per day, the time for loading and unloading at the terminals must be a minimum. As a corollary of this, speeding up the time of loading and unloading will decrease the labor required. The detention time at way-points should not be increased because of time required for loading or unloading the larger quantities of mail and cargo. This will require close cooperation between airplane designers, the airline operating officials, and those who operate the ground service at the airports, including the actual handling of cargo and mail in and out of the planes.

Mechanized equipment will be required, and in the selection of this equipment thought should be given to utilizing or adapting equipment which has been developed for somewhat similar work. From the outset, it should be determined to avoid perpetuating the pre-war handling-by-hand methods, as these were more flagrant than the flooring-and-rehandling methods used for mail and small freight by surface transportation companies. As to the mechanical means for expediting handling, two generic types of equipment will be required – containers and conveyors. Neither of these is new to the aviation industry, particularly the manufacturers. As applying containers and conveyors to airline use, the following detailed suggestions are extended:

Containers in the form of bags are familiar to airline operators through their use in air mail and air express. However, bags have such outstanding disadvantages that their use in the present manner cannot be continued and still gain the objectives desired. The disadvantages include the difficulty of moving a number of sacks at one time unless they are on a wheeled vehicle; they will not ride a roller conveyor; the size of a bag is definite and inflexible;

bags offer no protection to contents from crushing; loading bags with packages is a slow process; they require special racks to hold them open while being loaded. These disadvantages outweigh the advantages of being cheap, light, durable, readily repaired, and of occupying minimum space when empty.

One improvement that can be made on mail or cargo bags is to provide a stiff patch on one side of the sack so that it will ride a roller conveyor. The patent office, much to my disappointment, informs me this idea is not patentable, so it is given to you for what use you may make of it. The use of such a patch, however, would by no means overcome the other disadvantages in the use of bags.

Characteristics for the design and construction of airborne containers should cover:

Light weight, inexpensive, and reusable.

Able to resist rough handling; be readily repairable.

Strength to protect lading; cover to retard pilferage.

Shaped to facilitate quick loading with minimum of voids.

Should fold or nest in minimum space when empty.

Of such uniform size as best adapted for use on various planes; endeavor to have each carry approximately uniform weight of lading which might be 500 lb for exclusive cargo planes or less for combination planes.

Pliant cover to hold load in place and permit volume of lading to exceed rigid outline of container.

Self-attached means for quickly anchoring the container.

Should be designed for use in conjunction with means enabling one man readily to move loaded container.

The experience of Railway Express with containers backs some of the above conclusions. That company for many years has used wooden packing trunks by the thousand, also some of piano-box size on casters. Several years ago it standardized for new containers on a canvas container mounted on a spring-steel frame with skids below, which would ride a roller conveyor. These can be made so that they may be nested, when empty, like bushel baskets. The size found most useful for rail work was approximately 14 cu ft, slightly over 2 ft wide, 2 ft high, and 3 ft long. For air cargo, fitting this type of container with a pliant cover, with means for readily and quickly anchoring it to the floor of the airplane, and means for readily moving it within the airplane have all been worked out and are under process of further test and development. Other forms of container construction may encompass the desideratum previously outlined. It will be found that speed of loading, unloading, and securing cargo in the plane, moving the cargo to the appointed position dictated by the weight-balance required will be sufficiently expedited by the use of containers as to offset the tare weight thereof.

For conveyors, it is certain the magnesium roller conveyors, as introduced by Railway Express, will be found of utmost use both in expediting the handling and in lightening the labor required, as well as in decreasing the damage to shipments. The Jervis B. Webb Co. collaborated wholeheartedly in this development; and during the three years preceding the war, Railway Express equipped over 200 terminals with these magnesium conveyors, using in all about 27,000 linear feet. Railway Express before the war was probably the largest domestic user of extruded, heat-treated Dow shapes. With 18-in. rollers on 4-in. centers, supported by $3\frac{1}{2}$-in. channel side frames, these conveyors weigh approximately 7 lb to the linear foot. They are so light one man can instantly move them as the position of the lading changes. For airline work there is the added advantage that without serious weight penalty they may be airborne on the floor of the airplane with the cargo resting thereon in anchored light-weight containers.

Combined use of these two pieces of equipment – light-weight airborne containers, and light-weight airborne roller conveyors – provides means for one or two men, at a way-station, to unload the one or more containers bound for that station, then load into the plane, place at the position designated by the pilot, and anchor the containers which are put on at this way-station, within the time limitation for fuel and passengers. Placing mail sacks in these containers and thus handling several sacks as a unit expedites their movement. Eventually, a messenger will work in the container compartment of the airplane, transferring shipments between containers, then moving on the roller conveyor the container to be unloaded into position near the door for rapid discharge when the plane lands.

The use of roller conveyors as described above is applicable when the floor of the vehicle used to move the lading to the plane side is approximately the same height as the cargo door of the airplane. Accidental damage to the airplane and cargo doors will be decreased if the plane-side vehicles are not permitted to back up closer than several feet from the plane. The light-weight roller conveyors will be found a convenient method for bridging the gap between the tailgate of the vehicle and the plane, as well as for extending into the plane for handling cargo therein. When large volumes of cargo are to be handled, it will be found faster and more economical to use good-sized individually powered trucks or over-the-road trailers to move the lading to and from the plane side rather than a number of smaller units. The ability to drive a street truck to the plane side at practically any airport is an advantage of handling cargo by airplane that is not possible at most terminals where cargo is assorted and loaded on surface transportation vehicles.

When the cargo door of airplanes is elevated far above the ground, means must be provided for elevating the cargo. Of the several methods for elevating the cargo, three seem the most practical: namely, towing a portable elevator to the plane side and placing the lading on the elevator; a truck with a good-sized elevating body such as is used for re-icing refrigerator cars, which unit has the advantage of being able to transport a load to the plane side as well as elevate it; an inclined slat conveyor mounted on a light-weight truck chassis with power take-off, the height of the conveyor being adjustable at the upper end to suit the elevation of the cargo door and the lower elevation to suit the tailgate height of the feeding trucks. This unit can be provided with walkways on both sides and a light-weight roller extension into the plane itself. When large quantities are to be loaded and unloaded, the continuous-flow principle will handle more tons per hr than will be possible with the intermittent movement of either an elevator or an elevating truck.

The aviation industry as a whole has the opportunity now to work out the most efficient methods for cargo handling. But these can only be arrived at by a closely coordinated and combined effort of the airplane designers, the airline operating officials, the traffic men who are to obtain the lading, and engineers experienced in the physical handling of the type of traffic that is foreseen for the post-war period. A clean drawing board awaits the aeronautical members of the SAE for this development.

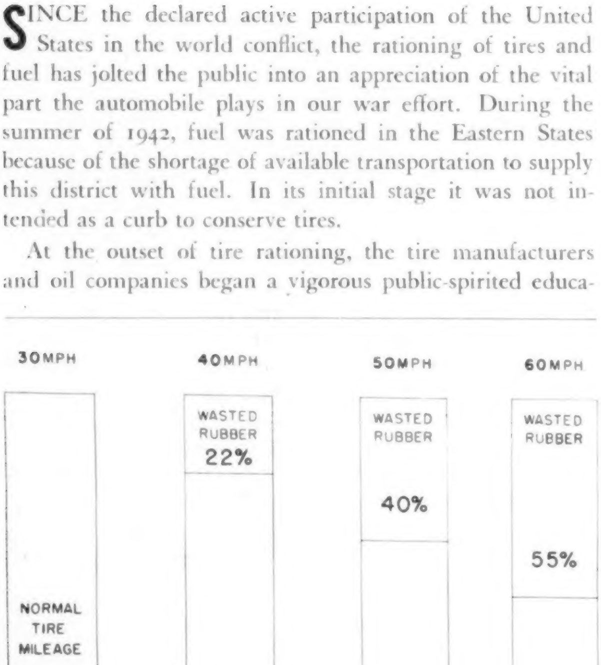
INCREASED ECONOMY with FU

SINCE the declared active participation of the United States in the world conflict, the rationing of tires and fuel has jolted the public into an appreciation of the vital part the automobile plays in our war effort. During the summer of 1942, fuel was rationed in the Eastern States because of the shortage of available transportation to supply this district with fuel. In its initial stage it was not intended as a curb to conserve tires.

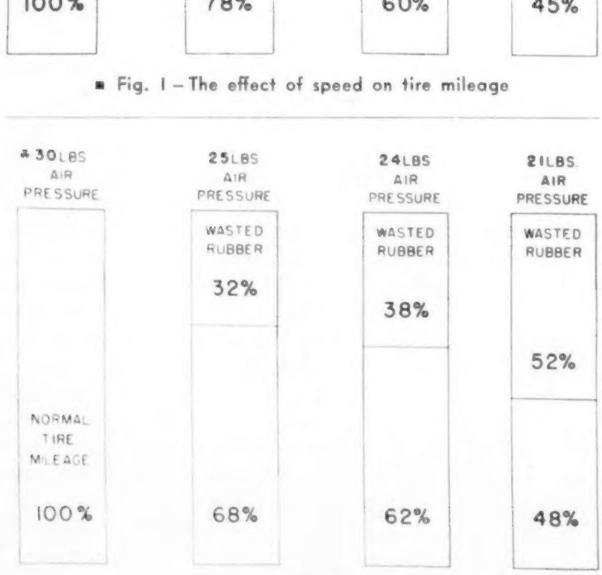
At the outset of tire rationing, the tire manufacturers and oil companies began a vigorous public-spirited educational program to preserve tires through proper use and servicing methods.

Figs. 1 and 2 indicate the reduction in tire life caused by under-inflation and car speed.

When fuel rationing materialized in the Eastern states due to the lack of available fuel transportation facilities, vocational schools and service stations cooperated in trying to increase fuel economy through correct car maintenance and driver education. This revealed that many of the cars driven by the general public were not operating as efficiently as they could be.



■ Fig. 1 – The effect of speed on tire mileage



* RECOMMENDED AIR PRESSURES VARY ACCORDING TO THE SIZE OF TIRE AND WEIGHT OF CAR 30LBS IS USED FOR ILLUSTRATIVE PURPOSES

■ Fig. 2 – The effect of tire inflation on tire mileage

THE present wartime rationing of gasoline has jolted those of us who drive cars into a realization that those cars, in most cases, are not being driven as efficiently as possible.

Out of this realization has come the desire to get as many miles per gallon from our rationed fuel as we possibly can, even at the cost of lower power, slower acceleration, and not quite as smooth a running engine as we have been accustomed to in the past. One of the suggestions that has been made for getting more miles per gallon out of our present engines is to "split" the engine, using only one-half of the cylinders for power.

This paper is a résumé of the results of a test program carried out by the Bendix Aviation Corp. to explore the possibilities of the split engine.

■ ■ ■

THE AUTHORS: EMIL O. WIRTH (M '36), chief engineer of the Military and Light Aircraft Carburetor Department of the Bendix Products Division, Bendix Aviation Corp., has been with Stromberg since 1925, and became chief engineer of automotive carburetors in 1935. Since the war this department has been developing and producing carburetors for tanks, auxiliary engines, armored vehicles and trainer airplanes. Mr. Wirth was co-author of a paper on "Highlights of Carburetion," presented before the SAE in 1938, and served on the CFR Motor Fuels Committee. For the past two years he has been vice-chairman of the Passenger Car Activity for the Chicago Section. ALBERT H. WINKLER, JR., is engineer in charge of research in the Military and Light Aircraft Carburetor Department of the Bendix Products Division. Mr. Winkler received a B.S. degree in mechanical engineering from Armour Institute of Technology in 1930 and a graduate degree of mechanical engineer in 1937 from the same school. He has been associated with the engineering and service phases of the Stromberg Carburetor since he joined Bendix Stromberg Carburetor Co. in 1930. Mr. Winkler delivered his first paper on carburetion before SAE in 1934, and since then has presented several other papers.

With good complete engine tune-up, the average mileage possible for the limited fuel supply available did not meet the driving requirements. This condition introduced the economical possibilities of a split engine or an engine in which only half of the cylinders are used for power.

[This paper was presented at a meeting of the Chicago Section of the SAE, Chicago, Ill., Oct. 13, 1942.]

[1] Chief engineer, Military and Light Aircraft Carburetor Department, Bendix Products Division, Bendix Aviation Corp.

[2] Research engineer, Military and Light Aircraft Carburetor Department, Bendix Products Division, Bendix Aviation Corp.

FUEL and TIRE RATIONING

• by EMIL O. WIRTH[1] and ALBERT H. WINKLER[2]

The Bendix Aviation Corp., being a manufacturer of an accessory that might be affected by this engine modification, formulated a test program to augment previous tests. This discussion will, therefore, be a résumé of our test results and some of the various problems confronting an individual when contemplating splitting an engine.

Theory of the Split Engine

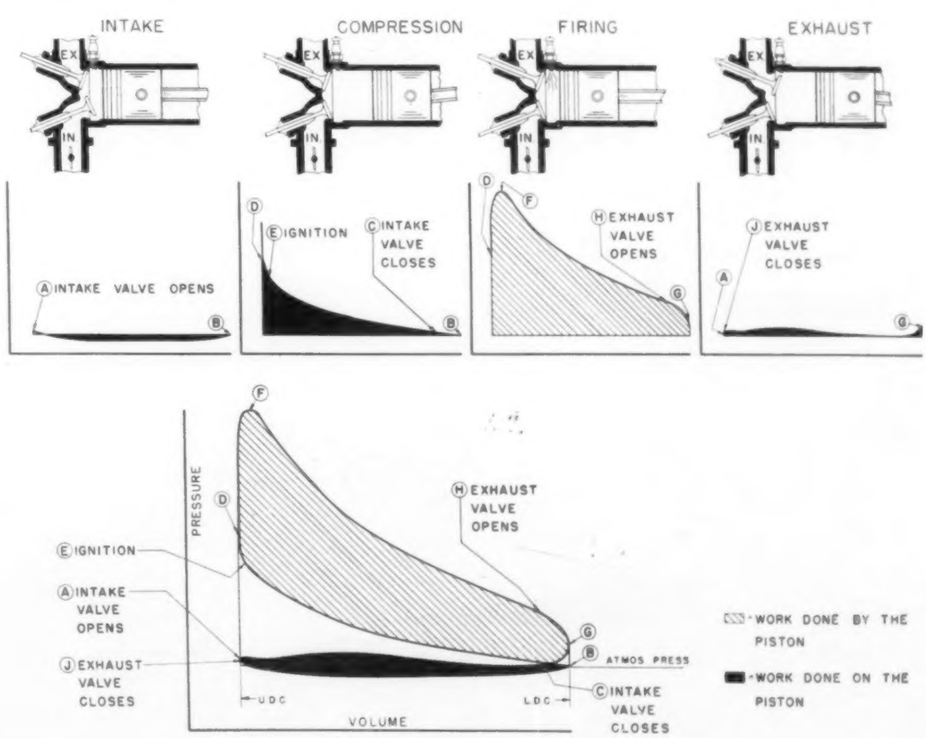
In operating a split engine, readjustments must be made in one's performance expectations and in one's driving technique. Gear shifting becomes more frequent and the road must be clear for a much greater distance before passing may be safely attempted. All these items become readily apparent when we consider that the available motive power has been reduced more than 50%, and the number of power impulses per revolution, halved.

In spite of all this, the normal operation of a split-engine car may be satisfactory to some drivers. Constant-speed operation above 15 mph is entirely satisfactory. The added economy obtained in conjunction with the expected limited fuel supply might overshadow the initial annoyance of the lowered performance factor.

Since the paramount factor in favor of the split-type engine is economy, how is it possible to obtain this added mileage from an engine that will have to develop the normal motive horsepower for a given constant-speed operation by merely eliminating a portion of the working cylinders of the engine? In order to explain this apparent paradox, it is necessary for us to review the various phases that compose the 4-cycle engine, with special reference to the indicator card which, to many of you, is quite understandable but to some may be only a hazy recollection.

Fig. 3 shows the four phases of the standard 4-cycle engine, intake, compression, firing, and exhaust strokes, and their respective general indicator cards. During the intake stroke, when the piston moves from the upper to the lower end of its stroke, air and fuel are drawn into the cylinder through, in this case, a carburetor with the throttle fully open. The mixture will flow into the cylinder with a small pressure drop between the outside air and the inside of the cylinder. The work done *on* the piston between points *A* and *B* is shown in black. The depth of the black area indicates the pressure drop inside the cylinder. The shape of this curve is a function of the engine speed and the length, shape, and size of the induction system for a given engine. The intake valve usually closes after BDC at *C* in order to take full advantage of the manifold ram and have maximum valve opening during a greater portion of the suction stroke.

When the piston starts up on the compression stroke, the



■ Fig. 3 – The phases of the standard 4-cycle engine with their respective general indicator cards and the complete indicator card – full-throttle operation

fuel-air mixture within the cylinder is compressed, requiring more work to be put *into* the piston in moving the piston from its lower position *B* to its upper position *D*. The black area under the curve therefore again indicates the work put *into* the piston. At *E*, the charge is ignited by the spark plug, and the pressure created by the burning gases raises the pressure to *F*. As the piston is forced down by the gas pressure within the cylinder, work is done *by* the piston, which is indicated by the shaded area under the curve *DG*. The exhaust valve opens at *H*, and there is a blow-down loss between points *H* and *G* as indicated by the change in the curve contour.

When the exhaust valve opens, the pressure within the cylinder is released and as the piston moves from its lower position *G* to the upper position *A*, the piston pushes or sweeps the gas from the cylinder through the exhaust valve. Inasmuch as the piston is pushing the gases out, work is being put into it. The amount is shown in black. The shape of this curve will be influenced by the engine speed and the size, shape, and construction of the exhaust system.

If we now combine all four diagrams into one and make a composite curve, it will look like the lower graph in Fig. 3, in which the work done *by* the piston is shown shaded and that done *on* the piston in black. This then is a full-throttle indicator card. The important thing to remember here is the relative shaded and black areas.

Fig. 4 illustrates the indicator diagrams of a cylinder operating under part-throttle, high manifold suction conditions. Under these conditions, the work done *on* the piston to draw a reduced charge into the cylinder is greater because the piston is drawing the charge through a restricted opening (partially closed throttle valve). This creates a higher vacuum above the piston than if the throttle were fully open as in the previous figure. The work done on the piston is shown in black between points *A* and *B*.

Since the pressure within the cylinder is lower than the outside pressure, the piston is drawn upward during part of the compression stroke by this pressure difference. The work done by the piston is shown shaded between the points *B* and *D*. After the pressure is equalized, work must be done *on* the piston to compress the charge above the outside air pressure. This portion of the stroke falls between *D* and *E* and is black, indicating work done *on* the piston.

Ignition occurs at *F* causing the charge to ignite and increase the cylinder pressure to point *G*. The gas pressure then pushes the piston down on the power stroke from *E* to *H*. The work done by the piston is shown shaded. At *J*, toward the lower end of the piston stroke, the exhaust valve opens, and on the up stroke sweeps the charge from the cylinder between points *H* and *A*. The work done *on* the piston is shown in black.

The lower diagram is a composite of the four cycles of operation, the shaded area representing useful work and the black area, work required to be put *into* the engine. As in the previous figure, the difference between the shaded and black areas represents the available work for operating a car. The important feature here is the fact that the black area almost equals the shaded area with little left over for available driving power. Since the work developed by the engine, either to overcome pumping losses, friction, or driving a car, is through the fuel, it becomes apparent that by reducing the pumping losses, a greater percentage of the fuel can be used for driving a car.

Fig. 5 shows two indicator cards having the same amount of available work for driving a car. This work is indicated by the amount of the shaded area after the black area has been subtracted from it. The one on the left is for an engine of approximately half the size of that on the right. Both produce the same available useful work, yet the total shaded area on the small-engine card is less than that on the large-engine card on the right. Hence, less fuel is required by the small engine to do the same work as the throttled large engine.

This graphically approaches what is accomplished in the mixture cycle when we cancel out part of the working cylinders in an engine to produce the so-called split engine.

Another item that contributes to better engine efficiency at low manifold operating condition is the smaller percentage of charge contamination caused by the exhaust gas flow-back during the valve overlap period.

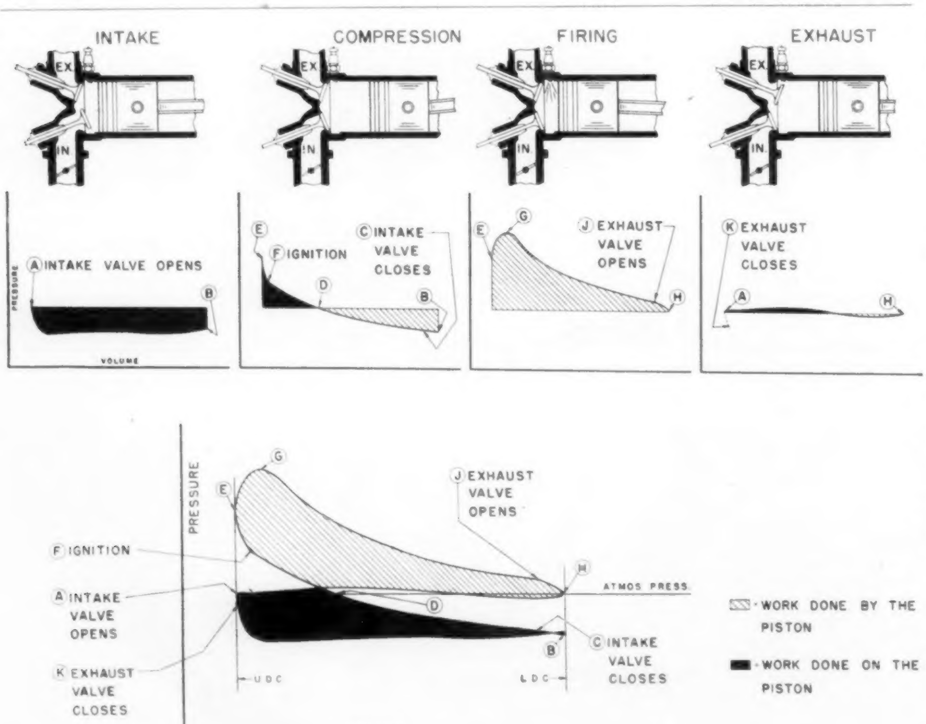
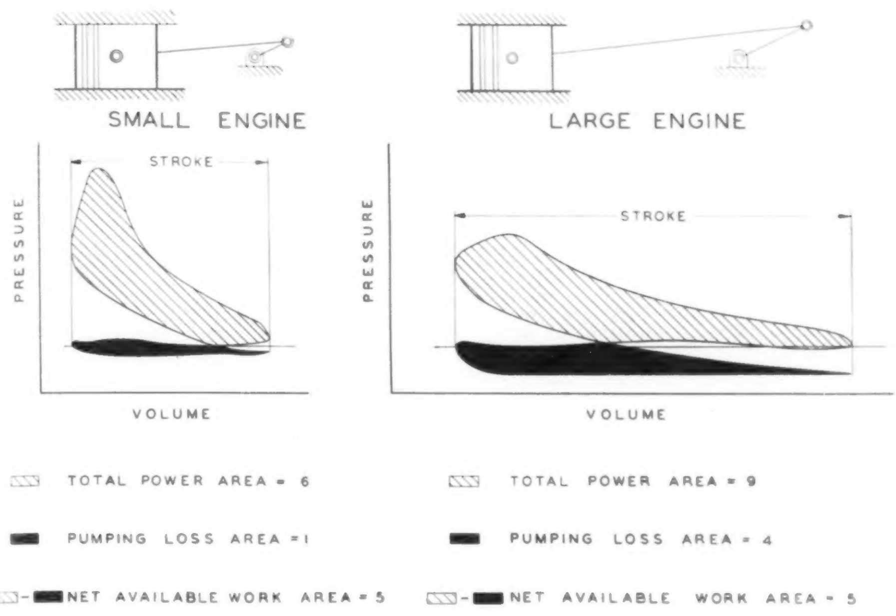
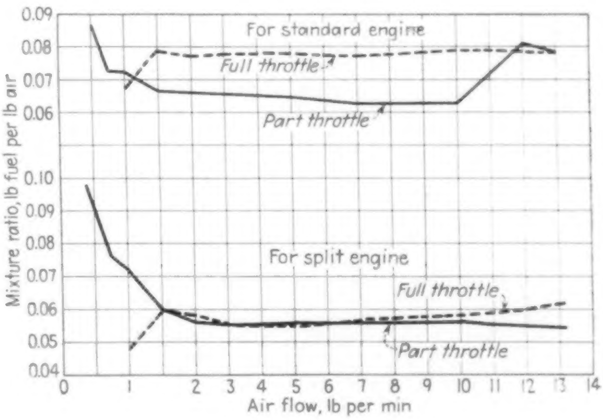


Fig. 4 – The phases of the standard 4-cycle engine with their respective general indicator cards and the complete indicator card – part-throttle operation



■ Fig. 5 (left) – Indicator card analysis

■ Fig. 6 (below) – Flow bench curves for standard and split engines



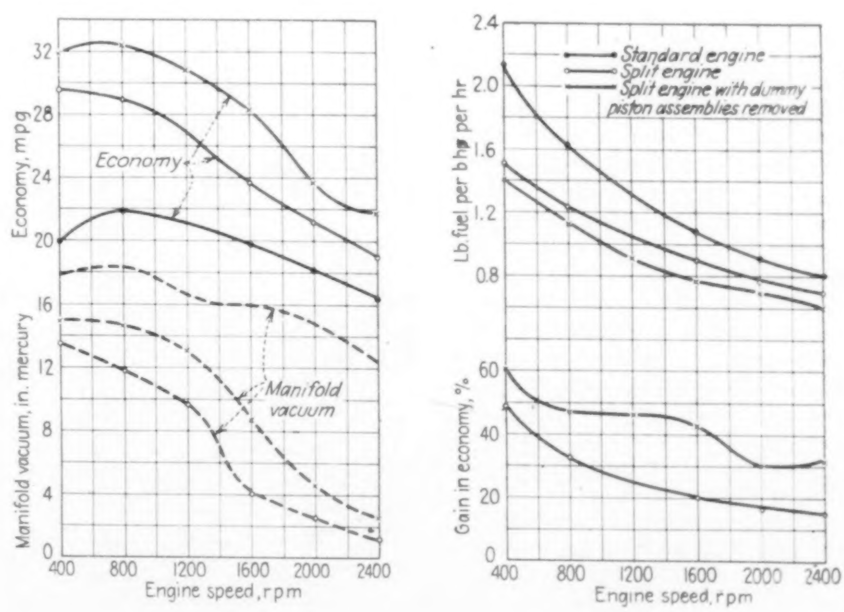
■ Dynamometer Test

A dynamometer test program was organized to check a popular-priced 6-cyl engine on the stand. This test included a full dynamometer run on the standard engine before modifying it to the split-type engine, of which the rear three cylinders were used for power.

Both full- and part-throttle fishhook curves were taken to determine the mixture requirements of the engine. Full- and part-throttle spark advance curves were then taken with the best mixture settings. The results indicated that a slight gain in part-throttle economy could be obtained if the spark setting was advanced approximately five deg from standard. No benefit could be had by changing the full-throttle spark setting from standard.

The pulsating air flow created by 3-cyl operation did affect the fuel flow and produce a rich condition. When the revised carburetor was rechecked in the flow bench, the full-throttle mixture curve practically coincided with the part-throttle curve. Fig. 6 shows the flow bench curves for both the standard and split engines. By blanking the power or bypass jet and using the standard engine main metering jet, the power mixture would be approximately correct, and the part-throttle economy only slightly affected.

Complete full- and part-throttle engine tests were then made with the best settings. The part-throttle results for a standard engine, split engine, and a split engine with the dummy piston assemblies removed are shown



■ Fig. 7 – Part-throttle engine characteristics for standard and split engines – standard 6-cyl engine, 216 cu in. displacement

in Fig. 7. Removing the pistons materially reduced the engine friction and eliminated the compression and expansion gas losses in these cylinders but resulted in rougher engine operation. Both the brake specific fuel consumption and the per cent gain in economy curves indicate the diminishing advantage in economy of the split engine as the load and speed are increased. A substantial economy gain at the lower engine speeds is present with the split engine for this type of test.

The corresponding full-throttle tests are shown in Fig. 8. While part-throttle operation favors the split engine, full-throttle operation discourages it. The available power output of the split engine is less than 50% of that for the standard engine, and the brake specific fuel consumption is greater than for the standard engine. Removing the dummy piston assemblies partially corrects this condition, but the increased roughness in engine operation would discourage this practice. The indicated horsepower of the split-type engine is approximately 50% of that for the standard engine.

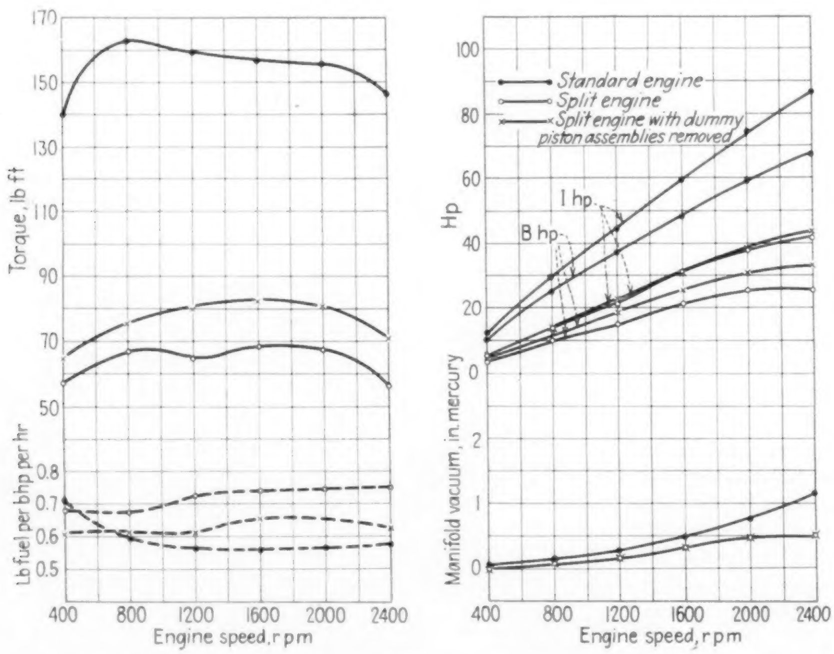
Fig. 9 indicates both the road load and full-throttle friction horsepower curves and the indicated specific fuel consumption. The full-throttle indicated specific fuel consumption curves again show the split engine to a disadvantage. An exhaust gas analysis did not indicate a greater spread in fuel distribution between the cylinders, but did record a richer mixture for each of the cylinders relative to the standard equipment. Fuel conditioning in the induction system due to the pulsations may account for this deviation since the indicated horsepower curves check reasonably well.

The part-throttle indicated specific fuel consumption curves are approximately equal over the probable operating range, indicating that the fuel conditioning process was not materially affected by the split-engine change-over.

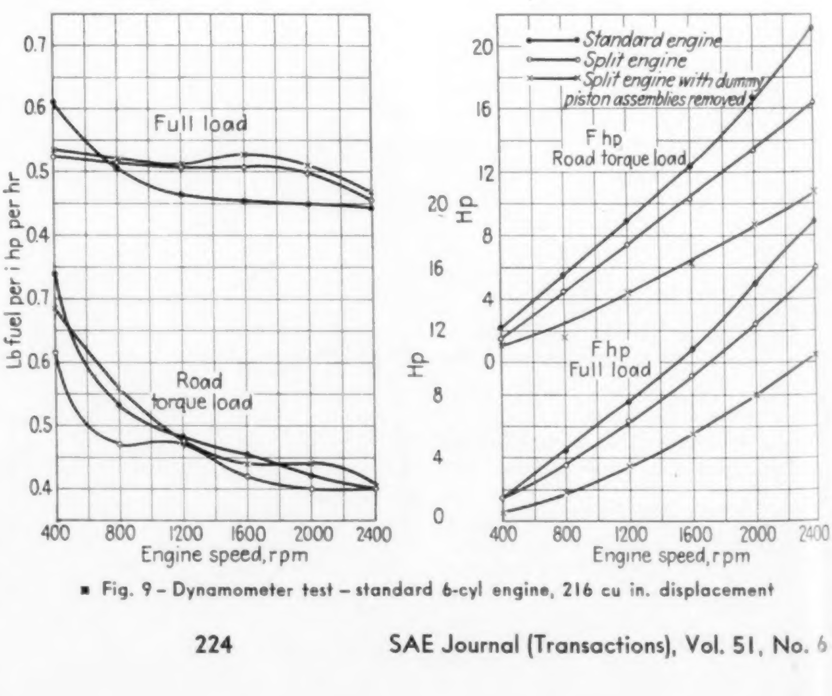
There is an appreciable drop in the available power at the rear wheels for acceleration when an engine is split. Fig. 10 shows the brake horsepower curves of the standard and split engine together with the probable road load horsepower. The area between the road load and the engine brake horsepower curves indicates the relative reserve power for hill climbing and acceleration.

An 8-cyl engine, equipped with two twin carburetors, was block tested. The rear carburetor was completely blanked off, the metering jets supplying the outer barrel of the front carburetor blanked, and the balance slot between the two barrels in the front carburetor eliminated. The above carburetor alteration did change the flow characteristics of the carburetor to a marked degree. The upper curve in Fig. 11 is the flow bench curve of a standard front carburetor. The curves located in the lower section of Fig. 11 are for the altered or special carburetor. The split-engine part-throttle mixture requirement curve is indicated by the full line in the lower graph. The approximate part-throttle air flows at 20 and 40 mph are likewise shown. Under the existing national speed limit of 35 mph, the part-throttle fuel-flow characteristics are permissible, as the fuel mixture ratio does not diminish for air flows up to 40 mph. It was necessary, however, to use one size larger main metering jet than standard for part-throttle operation.

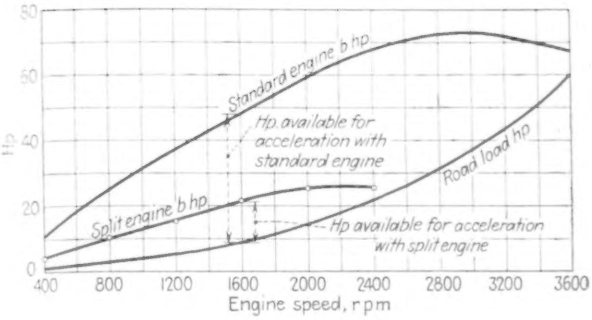
Two calibrations were necessary to obtain the full-throttle mixture requirements for the split-engine speed range due to the diminishing mixture ratios with increased air flows. The



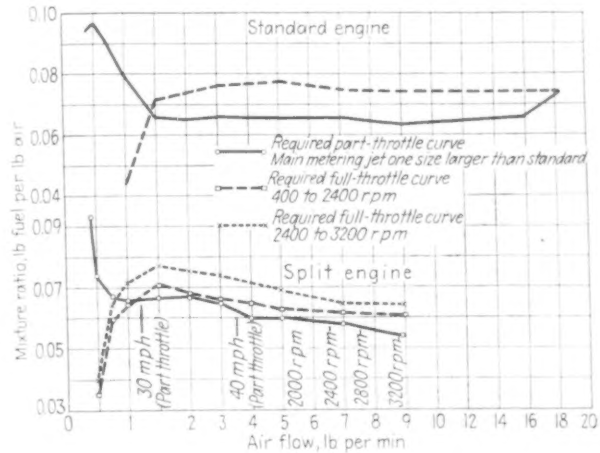
■ Fig. 8 – Full-throttle engine characteristics for standard and split engines – standard 6-cyl engine, 216 cu in. displacement



■ Fig. 9 – Dynamometer test – standard 6-cyl engine, 216 cu in. displacement



■ Fig. 10 – Acceleration characteristics for standard and split engines – standard 6-cyl engine, 216 cu in. displacement



■ Fig. 11 – Flow bench curves for standard and split engines

broken line curves in the lower graph of Fig. 11 represent the two full-throttle fuel curves together with the approximate full-throttle air flows for 2000, 2400, 2800, and 3200 rpm. If the two full-throttle mixture curves are blended together at the proper air flows, the full-throttle fuel-flow characteristics are quite conventional.

It is quite apparent that a compromise must be made, and the full-throttle mixtures will be richer than necessary during the lower speed range, and lean for the higher speed range. By making extensive changes in the carburetor, a single specification could be found that would result in good operation throughout both the part- and full-throttle speed ranges. Under the existing material and manufacturing shortages, this would hardly be possible.

The split-engine specific full-throttle fuel consumption checked fairly well with the standard engine; hence the leaner mixtures registered by the flow bench do not represent the actual fuel flows during engine operation. This discrepancy between the flow bench and actual engine operation is probably due to the pulsating air flows.

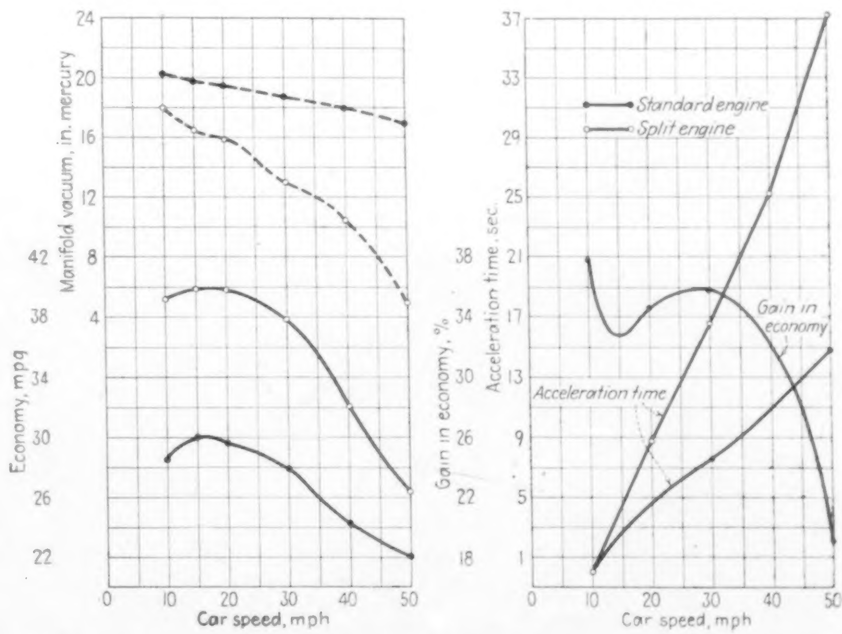
■ Road Test

Several different types of cars were road-tested to check the relative merits of the split engine and the standard engine. This test procedure also included changes that could be made in the carburetor to secure the maximum economy from a split engine.

The test results of two cars, one equipped with a 6- and the other an 8-cyl engine, will be discussed in this paper. Since we are comparing two types of units, the per cent change from the standard engine results and not the actual mileage figures should be the criterion. Fig. 12 shows the road load economy results of a standard 6-cyl car before and after splitting the engine. In this particular test, the carburetor was modified slightly to obtain the best possible fuel economy without a major carburetor change. The economy advantage of the split engine is rather impressive at the low car speeds but falls off abruptly when the car speed is increased above 40 mph. The time required to accelerate from 10 to 50 mph was approximately 250% of that for the standard engine. The split-engine acceleration at 10 mph in high gear is rough. When the carburetors were checked in the flow bench, the mixture requirements were practically identical with the dynamometer results on the same make engine as shown in the Fig. 6.

A further check was made on the split engine to compare the road economy results when the water temperature was increased from 140 to 185 F. The results of this test are shown in Fig. 13. The average gain in economy is approximately 12% and is somewhat higher than we have experienced under similar circumstances with standard engines. The data for Fig. 12 were obtained on a day when the temperature was 40 F, and the data for Fig. 13 on a day when the temperature was 70 F. This may account for the 6% loss in economy for speeds under 30 mph. The variation became less noticeable for speeds above 30 mph.

Fig. 14 shows the results obtained with a standard 8-cyl



■ Fig. 12 – Road test curves for standard and split engines – standard 6-cyl engine, 216 cu in. displacement

engine equipped with two twin carburetors. Three separate road tests were made as follows:

(a) Standard equipment.

(b) Split engine and no change in carburetor.

(c) Split engine with a solid gasket under the dummy carburetor making it completely inoperative.

Split engine operation is smoother when only one carburetor is functioning. When the fuel channels are blanked in the outer barrel of the front or operating carburetor, which ordinarily supplies the cylinders that are now dummies, it is necessary to increase the size of the metering jet one size to compensate for the fuel carry-over from the outer barrel. While the road economy was not materially affected by not blanking the outer barrel, the warm-up was seriously affected and caused excessive richness because of the increased fuel carry-over from the outer barrel when the choke valve was partially closed. The percentage gain in constant speed, part-throttle operation with the rear carburetor blanked out is shown in Fig. 14. This graph likewise indicates a diminishing per cent gain in economy as the car speed is increased.

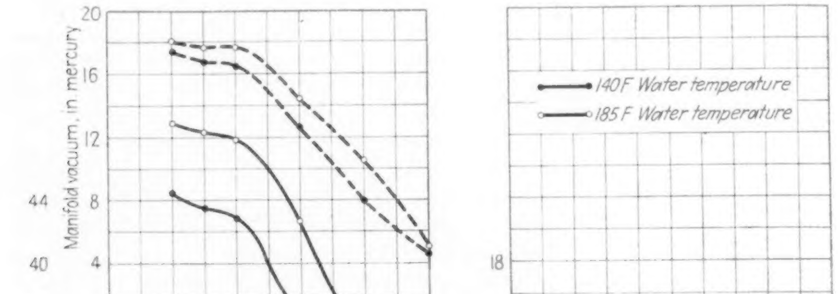
The acceleration characteristics of a split 8-cyl engine are not as objectionable as regards smoothness of operation as on a split 6-cyl engine, but the per cent increase in time to accelerate from 10 to 50 mph is still approximately 243% of standard. When accelerating from 10 to 30 mph, the time is reduced to 207% of standard.

Fig. 15 is a summation of the road economy tests of several standard makes of cars, indicating the per cent change in economy for a split engine relative to standard equipment. One set of data indicates the results obtained without changing the carburetor specification; and the other, when carburetor specifications are changed.

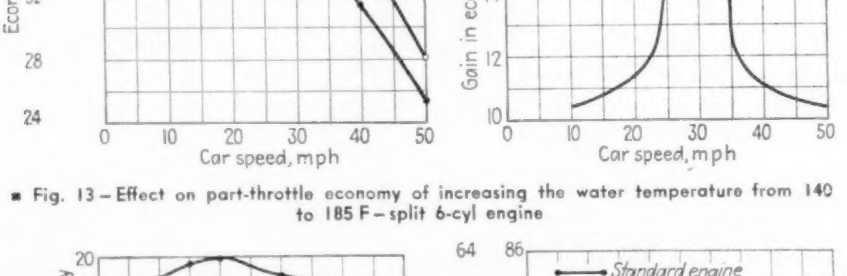
A similar set of data was obtained from another testing laboratory, and the results are summarized in Fig. 16. Averaging the curves, the indications are that a gain of approximately 20 to 25% would be possible for constant-speed operation under 35 mph. The traffic driving results indicate a possible gain of approximately 12 to 15%. If the carburetors are reworked, the approximate average gain would increase another 5 to 8% at constant-speed operation, and between 4 and 6% for traffic driving.

All of the road tests were conducted in temperatures above 40 F, and no actual cold-weather economy figures have as yet been made available.

In summing up the results, it would seem that the standard carburetor calibration in a number of installations would require only minor changes; while others, especially twin carburetors, would require a greater change.



■ Fig. 13 – Effect on part-throttle economy of increasing the water temperature from 140 to 185 F – split 6-cyl engine



■ Fig. 14 – Road test curves for standard and split engines – standard 8-cyl engine, 248 cu in. displacement

■ Low-Temperature Operation

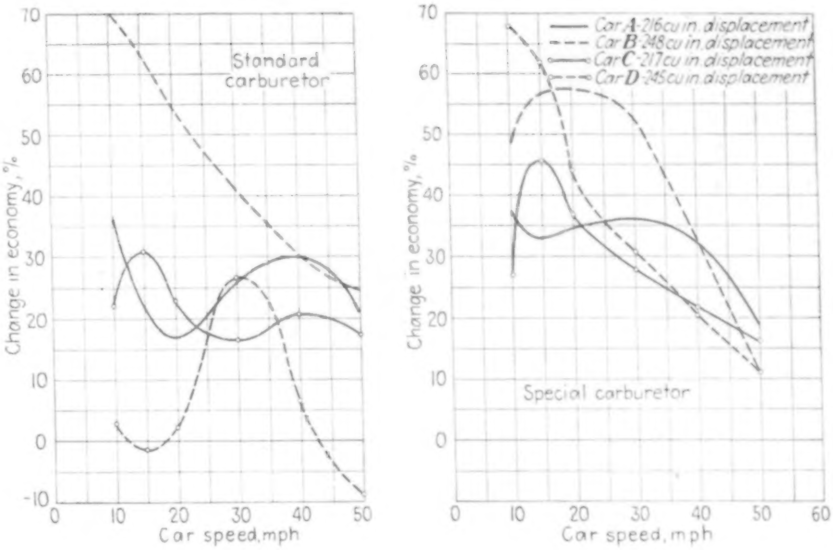
Operating a split engine in cold weather will create numerous problems that will have to be corrected by the use of materials on hand, more frequent adjustments on the previous automatic devices, and thorough driver education. Our limited work in the cold room has indicated that it is possible to start a split engine, and accelerate through the gears if certain precautions have been taken before the engine temperature has been lowered.

Crankcase oils must be as light and fluid as possible for low-temperature operation in order to ensure a start with minimum fuel consumption during both the start and warm-up periods. Standard automatic and semi-automatic chokes have been engineered in the past to start an engine reliably with cranking speeds ranging in the low thirties. To expect the

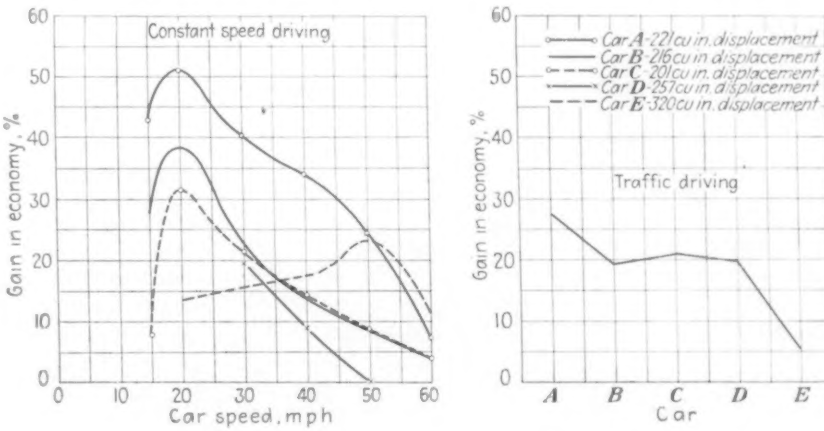
same starting ability from a split engine without being able to rework and redesign the starting elements is wishful thinking.

We have split an 8-cyl engine and successfully started it with a cranking speed of 36 rpm in three times the normal starting time. If the cranking speeds are maintained in the high fifties, this engine starts very well. Our experience with split 6-cyl engines has indicated to us that the starting problem is serious. Cranking speeds must be maintained in the seventies, and certain engine and carburetor adjustments seem to be advisable.

The possibility of starting a cold engine is much greater if the cranking speed is high and uniform. In the split 6-cyl engines that we have tested it was possible to maintain a higher and more uniform cranking speed if the three dead cylinders were piped together through their spark-plug openings. Venting the tube to atmosphere further increased the cranking speed but created a disagreeable noise, which could be eliminated by piping the outlet to the air cleaner and silencer. Before this change was made, the engine would hesitate periodically, almost stop, and then show a marked increase in speed. This fluctuation was almost certain to flood the engine before a successful start could be made. This tendency was not as objectionable on the 8-cyl engine. Venting or connecting the dummy cylinders through a restricted orifice, such as the spark-plug hole, will detract from the efficiency of the split engine.



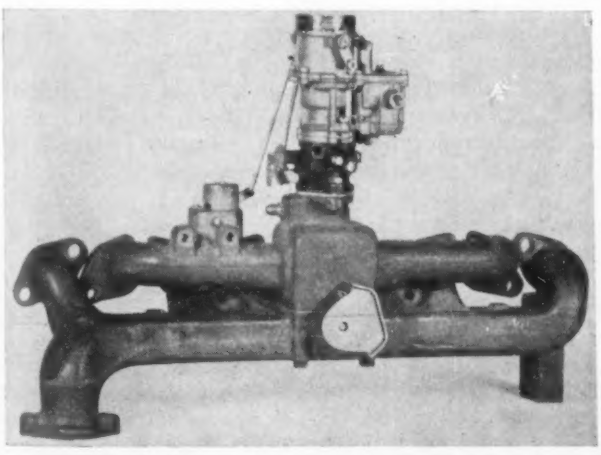
■ Fig. 15 – Economy road tests



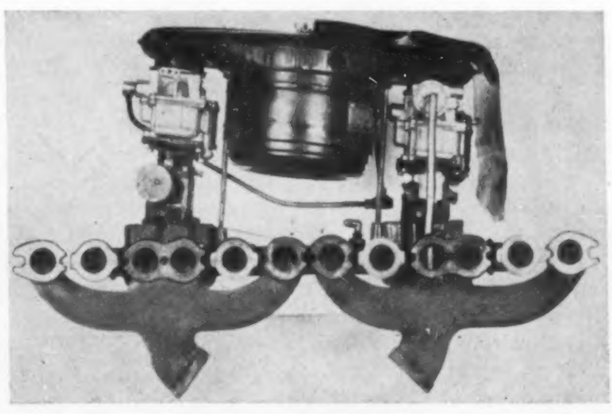
■ Fig. 16 – Economy road tests

The pulsations in a split 6-cyl engine, with a single carburetor and manifold, are rather severe. This condition seems to require a stronger poppet or thermostat spring action during the cranking period, as well as a reduction in the free air passages through the choke valve.

The one bright spot that materially helps the warm-up operation is the necessity for maintaining relatively high engine speeds throughout split-engine operation. The warm-up period depends upon the length of time it re-



■ Fig. 17 – One type of 6-cyl intake and exhaust manifold



■ Fig. 18 – Dual exhaust manifold

quires the engine and intake manifold to reach operating temperature. The cooling system is fixed, and for a split engine is oversized. This generally results in a lower operating temperature. Any modification, such as shields or baffles in front of the radiator, to decrease the warm-up period will be beneficial.

Fig. 17 shows a popular 6-cyl intake and exhaust manifold with the exhaust pipe outlet at the rear, and the thermostatically controlled exhaust heat valve at the center of the exhaust manifold. The normal slant of the intake manifold is to the rear, and the automatic choke is located on the exhaust manifold between the fourth and fifth cylinders. To obtain even power impulses in splitting the engine, either cylinders 1, 2 and 3; or 4, 5 and 6 can be used. If the rear cylinders are used for power, very little heat is transferred to the intake manifold, prolonging almost indefinitely the warm-up period. The rate of choke valve opening will parallel the normal opening rate with a much colder intake manifold. This results in a lean condition after the choke opens.

If the front cylinders are used for power, the exhaust gases will circulate through the intake manifold hot spot causing a heat transfer and a nearly normal intake mixing temperature. On the other hand, the rate at which the choke valve will open will be slightly slower than before; and the rear branch of the intake manifold will act as a sump for the heavy fuel ends. Of the two options, the latter seems the more successful.

When the exhaust outlet is located in the center of the manifold and adjacent to the heat valve, some of the problems are eliminated. However, on dual exhaust manifolds, as in Fig. 18, that were originally designed to carry the exhaust of four cylinders through each exhaust heat valve, the valve will pulsate when only two cylinders are discharged through it. When the heat valve pulsates, the maximum amount of exhaust gas heat that is available is not used to warm the intake manifold riser hot spot. Increasing the tension on the exhaust manifold heat valve 180 deg helped the warm-up on this type of equipment.

A large majority of the automatic chokes now in use employ a hot-air stove or tube on the manifold as a means of transferring engine temperature to the thermostat in the automatic choke. When either the volume of exhaust gas or volume flow characteristics are changed, it reflects in the heat transfer to the automatic choke. Referring to the manifold in Fig. 18, the front two cylinders were inoperative leaving only the two rear branches carrying hot exhaust gases. While the heat tube does run through the manifold, our tests indicated that the time required for the choke to open was approximately three times greater with the split engine. Insulating the heat tube and putting a shield in front of the choke housing and heat tube partly compensated for the loss in heat transfer at the stove.

The lower average manifold vacuum in the split engine is another factor that increases the difficulty of securing proper heat transfer to the automatic choke on this type of installation.

After the engine has been modified for the best possible starting and warm-up combination, the driver must still execute the required operations to move the vehicle. This is a decidedly difficult task and requires considerable jockeying of the clutch pedal to break the oil drag in the running gear. Once the oil drag is broken and high-gear operation is attained, the successive shifts are more easily made. Any additional external drag, such as snow, will further reduce the already low performance factor of the split engine.

■ Conclusions

Some of the items suggested, such as increasing either the normal water temperature or manifold heat, may not be practical on many cars because of the detonating problem during low manifold vacuum operation and the lowered octane number fuels on the market. They are items, however, to be considered.

The operation of the various automatic devices on the present cars, which perform definite operations when certain throttle positions and manifold vacuums are present, will no doubt have to be reworked and may be slightly modified. This will be another difficult problem to overcome in the split engine.

From the relatively limited tests that we have conducted, as well as those it has been our privilege to study, the advisability of converting to a split engine centers around the type of operation to be encountered. The straight-run economy results are very impressive, and the national maximum speed regulation of 35 mph, gas rationing, and the public response to slower acceleration, definitely favor the split engine as a means of traveling farther on a given quantity of fuel.

On the other hand, the maintenance cost for a split engine must surely be greater since shifting will be more frequent, and the practice of slipping the clutch to prevent the engine from stalling more prevalent. The wear and tear on the running gear, exhaust system, and accessories attached to the engine will probably be greater because of the added engine movement created by three-cylinder operation. In localities where the temperatures are low, the starting problem on some engines will definitely be serious. The fact that as the engine wear increases, the desirability of using heavier grades of oil to prevent oil consumption and maintain a reasonable oil pressure conflicts with good cold-engine starting, and this also may render the split engine impractical. The split engine, in the broad sense, has many arguments both for and against it under the existing emergency conditions, and they must be studied by the individual who is contemplating such a change. If the circumstances are favorable, the split engine would be adaptable; if not, it should not be attempted.

Previous to the national emergency, the split engine or a low-performance economy vehicle was not received too favorably by the public. Engineers were, therefore, trying to maintain the high performance factor of the present cars and yet reduce operating costs. Today, the field is considering a compromise, forfeiting performance and operating ease for greater mileage from the rationed quota of fuel. This in itself is a paradox for a great proportion of the country, since gas rationing is primarily a means of conserving rubber.

While engineers are reluctant to release anything but a finished product, and the split engine surely is not, compromises must be made and tolerated in order that the greatest number of people may benefit from any endeavor. It is with this thought that this paper was prepared, to expound as many sides of the problem as possible in the hope that a finished product might eventually emerge if the split-engine principle helps promote the public good.

STEEL CARTRIDGE CASES

by R. B. SCHENCK

Buick Motor Division,
General Motors Corp.

THE development of a practical method for producing steel cartridge cases became necessary when the critical shortage of copper made it imperative that the standard brass cases be replaced with ones made of steel.

A group of manufacturers, including Buick, worked on the problem with the Ordnance Department, and a cartridge case committee was formed, consisting of representatives from both Ordnance and industry. Very successful results have accrued from this cooperative program, and, according to latest reports, steel cartridge cases in practically all calibers will soon be in volume production.

This paper tells the story of how the process developed by the Buick engineers was accomplished, and describes the method in detail.

The Buick process has been applied successfully to the manufacture of steel cases of the 75-mm size, which are now being produced in large quantities.

Thirty-one major manufacturing operations are used in the production of these cases, the most interesting and unique ones being a series of four cold-drawing operations. These are carried out on a 750-ton double-acting press, where the cup is drawn from 6 in. to 15 in., and to the approximate form of the finished case, there being no substantial change in the diameter during the draw.

• • •

The Author: ROBERT B. SCHENCK (M '22) went to work for Buick when there were only 7½ million automobiles running in the United States, and alloy steels were in their swaddling clothes. He has had an active role ever since in the remarkable metallurgical developments of the past 17 years. Although born in Beacon, N. Y., he has been in sight of a steel mill ever since he went to Lehigh, from which he was graduated in 1909. He worked with Carnegie Steel at Homestead before moving into his first automotive connection with Weston Mott Co. Mr. Schenck is chief metallurgist in the Buick Motor Division, General Motors Corp.

THIS paper is a discussion of the Buick development of the steel cartridge case in the 75-mm size which is now being successfully manufactured in large quantities. It has been a very interesting and satisfactory research, having behind it the drive of necessity, since there had arisen as a result of the expanded armament program and isolation of certain sources of supply, a critical shortage of copper, the main constituent of brass from which practically all cartridge cases had previously been made. This critical situation was so accentuated as to make the development of a practical method for the production of steel cases in virtually all calibers an actual "must." Buick, together with a group of other manufacturers, worked closely with Ordnance on the problem, and a cartridge case committee was formed, consisting of representatives from both Ordnance and industry. Very successful results have accrued from this cooperative program, and, according to latest reports, steel cartridge cases in practically all calibers will soon be in volume production.

■ History

Before entering into a discussion of the metallurgical and production aspects of the Buick process, a brief outline of the history of the development may be of interest. The Buick project originated with a letter from the Navy Department in July, 1941, asking for ideas on the subject. As a result of this letter, work was started on a steel cartridge case.

At that time only bare details were available and we were in possession of no technical information as to metallurgical and physical requirements of steel cartridge cases. Military and Naval authorities had very little data on which to base specifications. The Buick management immediately provided funds for carrying on the work, although the project was not yet in the nature of a contract and Government funds were not immediately available for the purpose.

The original work was done on an extruding die about one-third the size required for a 5-in. Naval case. Preliminary efforts were mainly concerned with meeting dimensional specifications rather than physical properties.

The first extruding die worked satisfactorily, so experimental redraw and heading dies were made, and the first case was produced. The Buick management was favorably impressed with the sample case and submitted it to Naval authorities, who asked that development continue. Work was started on tools to carry a 3-in. steel cartridge case through the redraw operations. The 3-in. case was adopted because experimental work could be carried on faster with this than with the 5-in. size. While this work was in progress, other meetings were arranged with both Navy and Army Ordnance personnel. In November, 1941, the Army requested Buick to develop the 75-mm case, while the Navy decided to be guided by the Army developments. This was the beginning of the Buick program on the 75-mm steel cartridge case.

[This paper was presented by title at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 11, 1943.]

Mechanical Requirements

The purpose of a cartridge case is to act as a container for the explosive charge, to prevent the escape of gases resulting from the explosion, and to permit more rapid firing than would be possible without fixed ammunition. The wall of the case must be elastic enough to expand under the explosive pressure and make a tight seal against the breech wall of the gun. Unless it is properly supported by the breech wall, the pressure developed is sufficient to burst any case. A certain amount of clearance is necessary between the cartridge case and the breech wall to permit ease of loading and ejection under field conditions. The cartridge case must expand, first to take up this clearance, and continue to expand as the breech diameter increases due to the explosion pressure. The case must have enough elastic recovery to allow easy ejection after firing. In contrast to the high physical properties required in the body of the cartridge case, the mouth must be annealed to ensure complete obturation.

It is necessary to have a higher yield strength on a steel cartridge case than on brass because of the higher modulus of elasticity of steel. The modulus of brass is 14,000,000 against 30,000,000 for steel. The stress within the elastic range to produce a given amount of strain in brass is, therefore, approximately one-half that of steel. Other considerations are corrosion and sparking, which must be overcome by suitable protective coatings. The steel cases must be interchangeable with the brass and function equally well in service.

Metallurgical Requirements

The required physical properties of a steel case can be obtained either by cold-work alone or by quenching and tempering before or after cold-working. Cold-working alone involves the least number of operations, fewer shop problems, less equipment, and has been found adequate. Quenching after cold-working results in distortion of the case, which must be controlled by intricate quenching dies or by additional sizing operations. Quenching followed by cold-working eliminates the distortion problem and has been employed successfully on experimental lots. Where cold-working alone, or quenching before cold-working, is employed, a final stress relieving treatment should be used to improve further the physical properties.

In the selection of a material, a low carbon steel was chosen which could be cold-worked and stress relieved to the required physical properties without resorting to a spheroidizing treatment prior to cold forming. Among the compositions tried during the development were SAE 1015, 1016, and 1020, AISI C-1019, and a higher manganese type containing 0.20 carbon and 1.29 manganese. Experimental lots of C-1019 included coarse-grained, fine-grained, aluminum-killed and silicon-killed steels.

The composition finally adopted and now in production is an open hearth, high manganese, aluminum-killed, fine-grain carbon steel.

Of utmost importance is the physical quality of the raw material. Best mill practice with regard to discard, surface conditioning, and macrostructure is essential in producing suitable steel for cartridge cases.

As in all structural applications where stress is involved, the material in a cartridge case must possess a certain combination of strength and ductility for successful performance. In the side wall of the case, with the exception of the mouth, the yield strength must be high enough to prevent a degree of permanent set sufficient to cause difficult ejection. Also, the ductility must be adequate in all sections to prevent rupturing from the explosive pressure.

Manufacturing Operations

The following is a list of the major operations performed in the manufacture of the 75-mm steel cartridge case at Buick:

1. Cut off
2. Grind outside diameter
3. Heat
4. Extrude
5. Redraw hot
6. Size cold
7. Anneal
8. Pickle and rinse
9. Coin head
10. Bonderize
11. Draw
12. Trim end
13. First cold redraw
14. Second cold redraw
15. Third cold redraw
16. Trim end
17. Fourth cold redraw
18. Bonderize
19. Cold head
20. Flame anneal
21. First taper
22. Second taper
23. Flame anneal
24. Machine base
25. Face to length
26. Finish ream and counterbore
27. Inspect and repair
28. Draw
29. Phosphoric acid pickle
30. Paint
31. Bake

The appearance of the case at various stages of manufacture is shown photographically in Fig. 1.

It has been standard practice to start with the 3-in. blank to form the 75-mm case, but experiments indicated that by starting with a somewhat larger blank it was possible to get a better filling of the dies in the hot operations. Hence, the blank was increased from 3 to 3⅛ in. diameter, although this meant no essential change in the method.

The centerless grinding serves to provide a more perfect surface, eliminating defects which would tend to carry through the subsequent operations.

Heating for the hot cupping operation is done in a specially built induction heater, the piece being brought to temperature in 1½ min. See Fig. 2. The machine has two fixtures, which may be operated simultaneously, permitting a production rate of 80 pieces per hr. During the

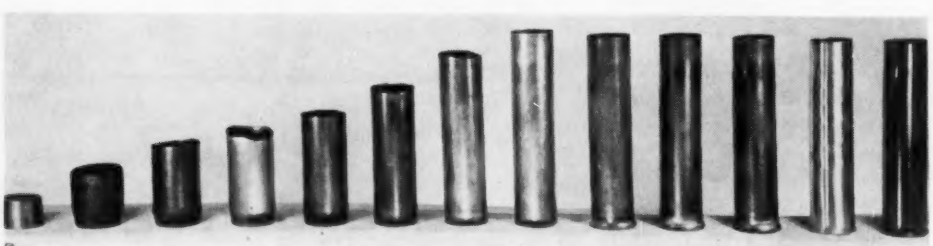
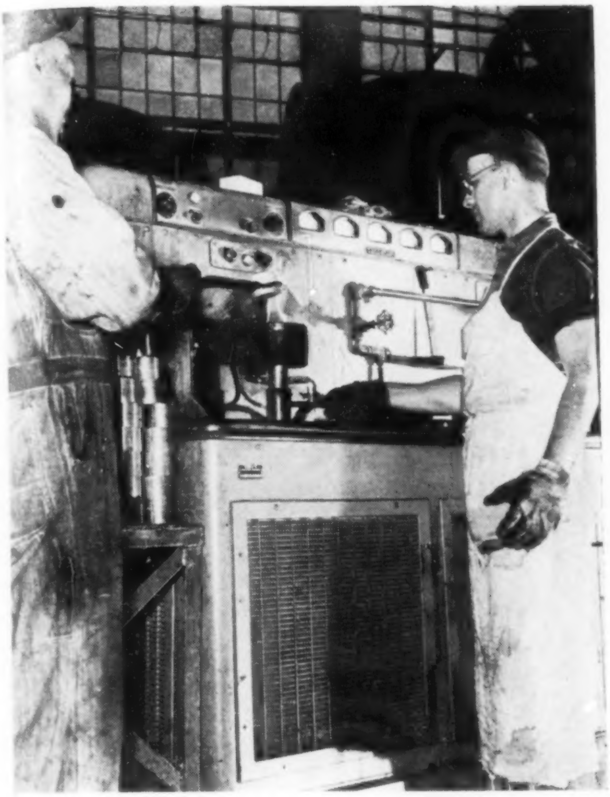


Fig. 1 – The appearance of the case at various stages of manufacture



■ Fig. 2 – Induction heater for hot extrusion and redraw

brief period of heating, a yellow gas flame is directed onto the steel, enveloping it and preventing scaling. The current kicks off when the piece has been brought to temperature, and it is then transferred to a gas-fired muffle furnace where it is held for 7 min to ensure an even temperature throughout the steel. This set-up is only temporary. Two new induction heating units are on order which will eliminate the soaking furnace.

The extrusion is done in a standard forging press of the crank type common to many automotive shops. See

■ Fig. 3 – Hot extrusion, as done in a standard forging press of the crank type

Fig. 3. On this press, the upper die member is a hot die steel punch, nitrided for improved wearing qualities and rounded at the end to form a smooth radius at the base of the cup. The lower die has a cavity the size of the formed cup, or virtually the same diameter as the heated blank. The punch extrudes the blank from 1¾ in. to about 4 in. A standard forging grease or graphite paste is used to lubricate the dies, and there is no trouble with sticking of the formed piece because a stripper on the punch removes the part on the upstroke of the punch.

In the early stages of the work, only one hot cupping operation was used, but it was later determined that better results could be obtained if a second hot operation was carried out after the piece had cooled somewhat. See Fig. 4.

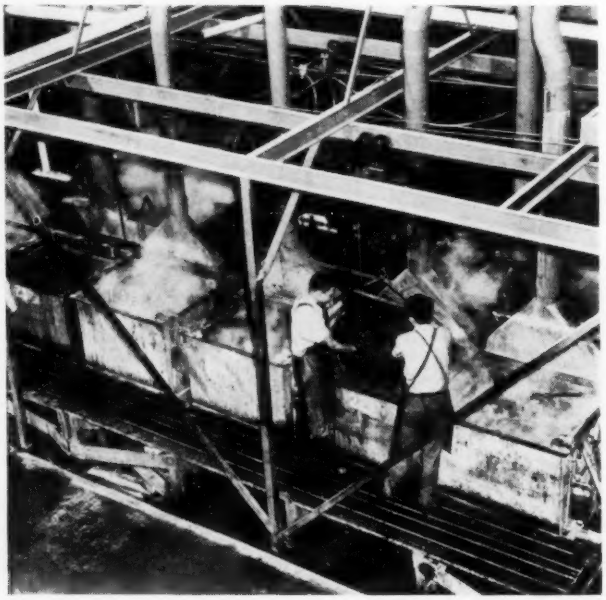
■ Fig. 4 – Hot redraw

The second hot press is the same as the first, except for slightly altered tooling extending the draw 2 in. further, or to about 6 in.

After the original set-up was worked out, an extra operation was added to control the wall thickness. The cup is cold sized in a press, drawing it out only slightly. This is done after the hot redrawn cup has cooled in air.

After annealing, followed by acid pickling and washing to remove scale, the head of the cup is coined in a crank press. This operation sizes it accurately as well as giving this portion of the case some extra cold-work, because in the subsequent cold-drawing operations the head does not receive as much work as the walls. Bonderite treatments serve both to clean the surface of the case thoroughly and to etch it slightly so that minute pockets will retain drawing compound in the cold operations. See Fig. 5. Further, the thin surface layer of zinc phosphate deposited in this chemical treatment acts as a lubricant during cold drawing.

Hydrogen embrittlement resulting from the acid pickling and bonderite operations tends to cause breakage during

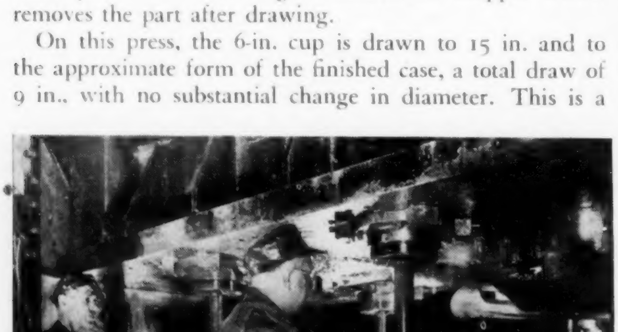


■ Fig. 5 – Bonderizing treatment

cold drawing. This trouble is eliminated by annealing, which drives off absorbed hydrogen.

Perhaps the most interesting step in the entire operation is the series of four cold-drawing operations, all carried out on a single 750-ton Clearing double-acting press. See Fig. 6. The four punch-and-die stations are placed to form the four corners of a rectangle, well within the normal platen area, and are so arranged as to carry approximately an equal load, of somewhere near 150 tons per station. The depth of draw varies slightly over the four dies, being controlled by the length of the punch. The cartridge case progresses from the left-hand die to the right-hand die on one side of the press, and is then handed through the die opening and placed on the die directly across from the second station, finally moving to the right-hand die on that side. Four operators handle the press, one at each die. Each punch has an integral mechanical stripper which removes the part after drawing.

On this press, the 6-in. cup is drawn to 15 in. and to the approximate form of the finished case, a total draw of 9 in., with no substantial change in diameter. This is a



■ Fig. 6 – Close-up of the 750-ton double-acting press for the four cold-drawing operations

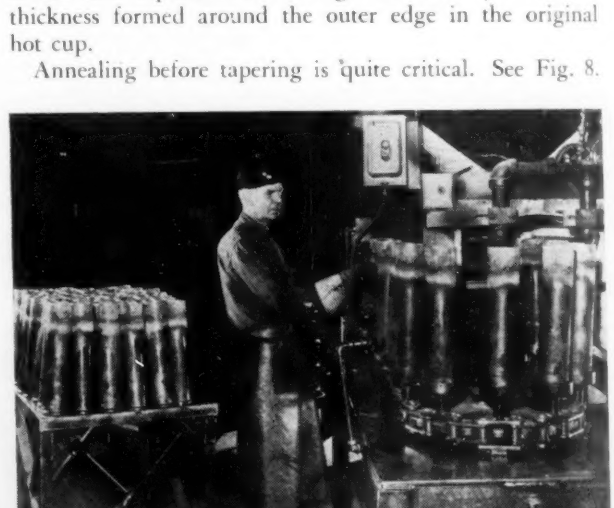
real test of the drawing qualities of the steel, and it is noteworthy that few pieces are torn or split in these severe operations. A special drawing compound was developed, the base material of which is machine oil.

The case is cold headed in a press equipped with a two-stage indexing die, the lower die being built so that after the first stroke it can be moved across the press, bringing the second die impression under the punch carrying the case. See Fig. 7. In these operations, the steel in the base

■ Fig. 7 – Cold-heading operation, as done in a press equipped with a two-stage indexing die

is flared out so that the flange can be machined, the steel to fill this expanded rim being furnished by the extra thickness formed around the outer edge in the original hot cup.

Annealing before tapering is quite critical. See Fig. 8.



■ Fig. 8 – Mouth annealing

It is limited at the mouth end to a depth of about 2 in. to prevent splits during tapering. The temperature is closely controlled. In this operation, the cases are mounted on a conveyor and moved between two rows of $2\frac{1}{2}$-in. radiant gas burners. These are mounted in a furnace horizontally at the proper height to concentrate the heat on the mouths of the cases as they travel through the furnace. The cases are rotated slowly as they move past the burners.

Tapering operations are perhaps the most critical of all steps since, in these two press operations, the cold steel must be made to flow into the desired taper without wrinkling or distortion and without the support of a punch on the inside. The case is simply forced up into a tapered die cavity in two stages, both on the same press. The second stage does have a punch which extends about 3½ in. into the case. This is required not for the overall taper but to support the metal in forming the reduced section at the mouth.

Punches used on all the cold-drawing and tapering operations are made of hardened high-speed steel chromium plated. The plating improves the wearing qualities and provides better antifrictional properties. Lower dies in the cold operations are generally steel rings with tungsten carbide inserts on the working surfaces.

After the tapering operations, the mouth end becomes appreciably harder because of the cold-working, so it is annealed once more, this time to a depth of about 3½ in. from the open end. This softens up the mouth and assures complete obturation in firing, that is, a tight seal in the gun barrel to prevent the explosion from blowing back into the breech.

The cases are inspected and transferred to automatic lathes, which face the head, rough form the flange, finish form the flange, and drill the primer hole. These lathes have five stations, four for the machining operations and one for loading and unloading. Reaming and counterboring the primer hole are performed on vertical machines, and great care must be exercised in these operations since specified tolerances are unusually close.

Two of the machining operations that have to be performed on the cartridge cases are shown in Figs. 9 and 10.

Final stress relieving is performed in a batch-type electric furnace accommodating 316 cases. This heat-treatment adds roughly 10,000 psi to the yield and ultimate strengths of the case.

The coating specified to provide protection against corrosion and sparking is an unpigmented baked phenolic varnish. Before painting, the cases are given a phosphoric acid pickle, which provides a bond for the paint and also some additional protection against corrosion.

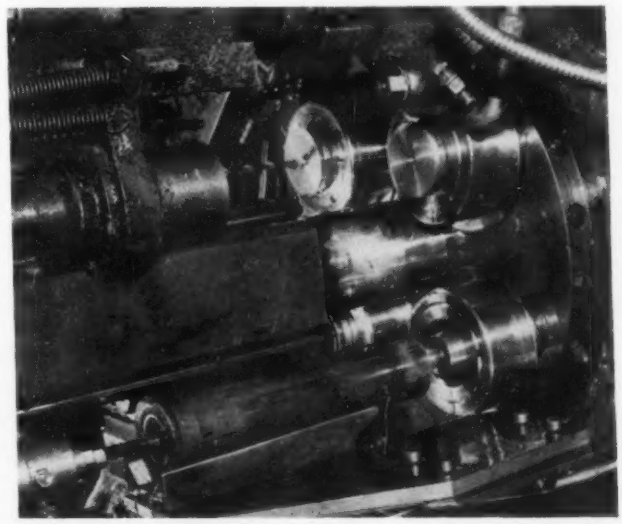
The painting and baking operations are fully automatic, the cases being mounted vertically on a conveyor with the open end up. They rest on fixtures attached to the conveyor chain so that as they pass the spray nozzles they can be spun by a motor-driven rubber belt. One spray nozzle is mounted on a traveling arm which descends into the case and lifts out at a uniform speed while the nozzle directs a spray of varnish over the rotating surface. At the same time, outside nozzles are positioned to coat the base and wall thoroughly and uniformly.

Once coated, the cases are carried slowly between two banks of infra-red lamps, 64 on each side. The conveyor loops twice at the end of the lamp bank so that the cases travel three times through the baking zone requiring 48 min in a temperature approximating 360 F. When they emerge at the opposite end from the painting station, they pass through an exhaust cooling hood and out to an unloading station where they are thoroughly dry and cool enough to handle.

The inspection of the 75-mm steel cartridge case starts with receipt of the raw material and ends with the final acceptance tests required by Ordnance. This inspection can be divided into three classifications, namely: raw material inspection, process inspection, and final inspection for acceptance. Raw material inspection includes determination of chemical composition, grain size, inclusion rating, macrostructure, and physical condition of the bar stock with respect to mechanical defects. Process inspection includes numerous dimensional checks, hardness tests, and visual examination for physical defects which may occur in processing. Inspection for final acceptance by Ordnance covers hardness, micro- and macro-examination, ductility, corrosion and abrasion resistance, and ballistic properties.

One unusual type of inspection tool has been developed to permit close observation of the interior wall of the case. A cone-shaped piece of steel slightly smaller in diameter than the case is chromium plated to a mirror finish on the outer surface of the cone. A wire is attached to the apex of the cone and a small light arranged to illuminate the interior of the case and the bright surface of the cone. The unit is lowered to the base of a case and drawn slowly upward while the inspector watches the mirror surface. The inner surface of the case is reflected in magnified form so that any surface defects are readily observed.

■ Fig. 9 – Close-up of the V. & O. trimmer for automatic trimming of cases (operation 12)



■ Fig. 10 – Close-up of the automatic lathe for machining the base, with the case in position

AIRCRAFT ACCESSORY S

THE accessory system in aircraft is used to convert mechanical energy at the aircraft engine to another form, transmit this energy to points throughout the airplane, and do an accessory task. The accessories are used to increase the safety of flying, the ease of handling the airplane, the aerodynamic efficiency of the airplane and the comfort of personnel.

In a properly designed airplane none of the accessories are essential to flight. The de-icing system may be noted as an accessory which improves the safety of the airplane. The automatic pilot increases the ease of handling the airplane. Examples of accessories which increase the aerodynamic efficiency are: landing gear retraction and flap actuating mechanisms, altitude controls, and propeller pitch controls. The comfort of personnel is improved by such accessories as heating devices, lighting equipment, and cooking facilities.

The accessories which are used can be grouped as follows:

a. Aircraft.
b. Engine.
c. Personnel.
d. Lighting.
e. Radio.

A list of accessories would include more than 100 items, some of which are small in so far as power consumed is required. The magnitude of this list gives an estimate of the accessory problem and indicates that it has considerable importance.

The weight of accessories and the penalty involved in their use is difficult to evaluate exactly in many cases. A certain portion of retractable landing gear weight is chargeable as an accessory since the airplane could fly successfully with a fixed landing gear. It is difficult to determine the proportion which is charged as fixed equipment and that which is accessory equipment. The same condition applies to a controllable pitch propeller, where it is difficult to determine the difference between a fixed pitch propeller which would permit the airplane to fly and the controllable pitch propeller. However, an analysis of several airplane types indicates that the accessory equipment equals approximately one-third the useful load of the airplane. A gain which can be made in this weight is reflected in an increase of useful load.

Several types of accessory power have been used in aircraft. In all cases, the main engine or engines are the basic source of power. It is the problem in the accessory system to install a generator on the engine, to transmit the power developed by this generator, and to use the power at the proper point in the airplane. Since power must be taken from the engine, certain limitations are imposed on any generator which is used thereon. In the first place, space on the accessory section of a high-output aircraft engine is very limited. It is necessary to install as many as eight large accessories on the rear section of a radial engine. This means that quite small envelopes must be assigned to each accessory, and the manufacturers thereof must utilize considerable ingenuity to avoid overlapping the territory assigned to others. With the increasing demand for accessory loads, a new factor has recently entered this picture. It is likely that certain accessories will exceed the torque allowances designed into the engine. This means a redesign of internal gearing within the engine accessory section.

Another factor which has disturbed generator manufacturers is the limitation with respect to overhung moment. Most radial engines having a rating of more than 1000 hp can successfully carry a generator whose moment is less than 275 in – lb. Development of larger accessories will increase this moment arm until it is necessary to carry an overhung moment of approximately 500 in – lb on a mounting circle whose diameter is only 5 in. This load imposes severe stresses on an accessory section, and indicates that in the future design procedures for the rear sections must be considerably altered.

The future with respect to space on the accessory section of the main engines is not bright. Engine designers are developing methods of increasing the power which can be taken from the main engine without increasing the area of the rear section. In fact, this area tends to decrease with increased power. Accessory sections which are already too crowded for convenient use are quite likely to become examples of impossible conditions.

The types of accessory power which have been utilized are:

a. Mechanical (direct take-off from the engine).
b. Hydraulic.
c. Pneumatic.
d. Electric.
e. Combinations thereof (electric-motor-driven hydraulic pump).

The requirements of a successful accessory system may be described as follows:

a. Reliability.
b. Light fixed weight.
c. High efficiency.
d. Simple installation and maintenance.
e. Reasonable cost.
f. Low vulnerability.
g. Ability to duplicate accessory power on the ground.

The importance of light fixed weight is best shown in the case of starters and storage batteries. These items are used only on the ground, and any weight imposed by these units is a handicap to the airplane while in flight. The efficiency of the accessory system is also extremely important because it determines the amount of fuel which is consumed during flight. Aeronautical engineers are quite likely to forget this factor. Certain types of accessories

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

SYSTEM

by LT.-COL. T. B. HOLLIDAY
U. S. Army Air Forces

which have extremely low fixed weight will actually consume many times their own weight in fuel during the course of a 5- or 10-hr flight. This weight should also be charged to the accessory. An example of this case is the mechanism for actuation of landing gear and flaps. If the units cause fuel to be consumed during flight the weight of that fuel should be charged to the fixed weight of the equipment.

An accessory must be reliable to justify its installation in aircraft. The factor of reliability should be considered at all possible operating temperatures and altitudes. If possible, the accessory generator and transmission lines should be duplicated so that an emergency condition cannot arise in flight.

The factor of vulnerability may be considered as a portion of reliability in so far as military aircraft is concerned. To achieve this end, the accessory system should automatically protect itself from gunfire wounds.

Comparison of the four types of accessory power immediately shows that each has certain advantages. Direct mechanical take-off of power is the lightest in weight and the most efficient. However, this means that the driven accessories must be at the engine when many of the accessory loads are far removed therefrom. In addition, the number of accessories which can be located on the engine are quite limited, as noted above. Therefore, this type of accessory power is extremely limited in scope and cannot be considered for most of the accessory items.

Hydraulic power has light fixed weight and is ideal for accessories which require a straight-line movement. This type of power is not very efficient and is weak with respect to vulnerability.

Pneumatic power has many of the same characteristics as the hydraulic, although it uses a gas instead of a liquid. As a result, it is slower in operation, is less efficient, and is particularly weak at altitude since the amount of fluid is decreased.

Electric power is not as light in fixed weight as hydraulic power, in the case of generators and motors. For equal efficiency, its weight in transmission lines is less. It is considerably more efficient, is much less vulnerable to gunfire, and connection to ground sources of power is more easily made.

Practically all of the various accessories have used electric power at one time or another with the exception of wheel brakes. This has been one problem which the electrical engineer has not been able to solve and which has been done in a superior fashion by hydraulic power. On the other hand, the hydraulic engineer has not yet been able to operate radio equipment. It is quite likely that the electrical engineer will find a suitable answer for the braking problem before the hydraulic engineer has accomplished a solution for the radio problem. In fact, a potential solution for the electric brake problem is being developed.

It is the function of the accessory power generator to convert mechanical power at the engine to another form. This generator develops pressure whether it is measured in pounds per square inch or in volts. The discussion which follows will be limited to hydraulic and electric power, since these two have found the greatest application in aircraft.

Hydraulic generators are pumps which may be described as vane, gear, or piston types. A typical pump, which is used at 1000 psi pressure, is shown in Fig. 1, and its performance curve by Fig. 2. The weight and power

---

ACCESSORY items are defined by Col. Holliday as that equipment not essential to flight. They are the items added to make the airplane easier to handle and safer to operate.

One simple and efficient way of obtaining the power to drive the accessories is by direct mechanical take-off from the engine. Use of this method is restricted because of space limitations in the rear of the engine where the unit must be located.

The next method considered is the use of hydraulic power. Although it has a lighter fixed weight, as compared with electrical units, there are disadvantages such as high loss in efficiency in cold weather. A pneumatic system – also possible – is not satisfactory because of a definite control lag.

The final system discussed by Col. Holliday is the electrical one. Its one chief disadvantage, that of excessive weight, is being rapidly overcome, making it seem as if the use of electrical power is definitely the coming thing.

In the matter of transmission, the electrical is far more efficient than the hydraulic system – and this advantage becomes even more apparent as the length of the transmission lines increases. Electric wires are also quite invulnerable.

Voltage and its hydraulic counterpart – pressure – must also be considered. A standard of 24 v is now being used, a figure that can be increased almost without limit, whereas hydraulic pressures already seem to be approaching their upper limits.

■ ■ ■

THE AUTHOR: LT.-COL. T. B. HOLLIDAY (SM '41) graduated from Purdue University with the degrees of bachelor of science in electrical engineering (1927), master of science in electrical engineering (1929), and electrical engineer (1931). He was assistant instructor in physics at Purdue University from 1927 to 1929, at which time he entered Wright Field as a junior electrical engineer in design and testing of aircraft lighting. In 1933 he engaged in a study of methods to reduce radio interference, and later supervised the development of a-c power supply and equipment for use in aircraft. He became assistant electrical engineer in 1936, and associate electrical engineer in 1940.

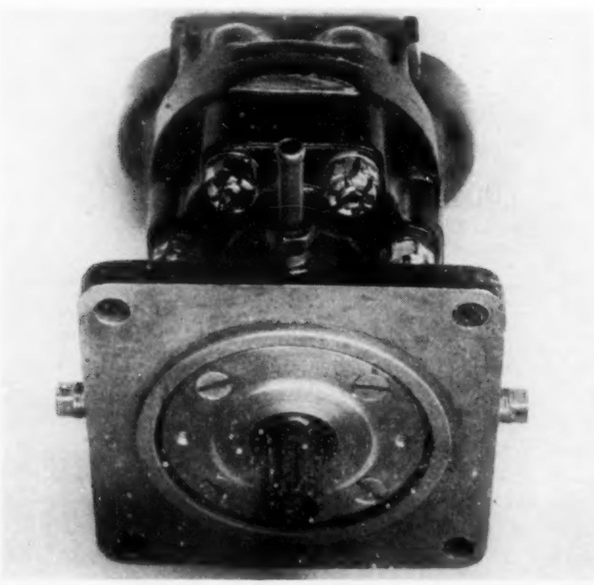
---

characteristics of hydraulic pumps are shown by Fig. 3. The development of the "live line" type of pump marked the greatest advancement in hydraulic pumps in so far as efficiency was concerned. Other types of pumps operated at full load continuously by discharging the fluid output through a pressure regulator release valve. This meant that the fuel consumed by the pump was a maximum throughout the flight, even though the pump was required for only a short interval at the beginning and end of the flight. This fuel consumed should properly be charged to the weight of the pump. The "live line" development makes a hydraulic pump more like an electric generator since it maintains pressure without imposing high loads on the engine drive.

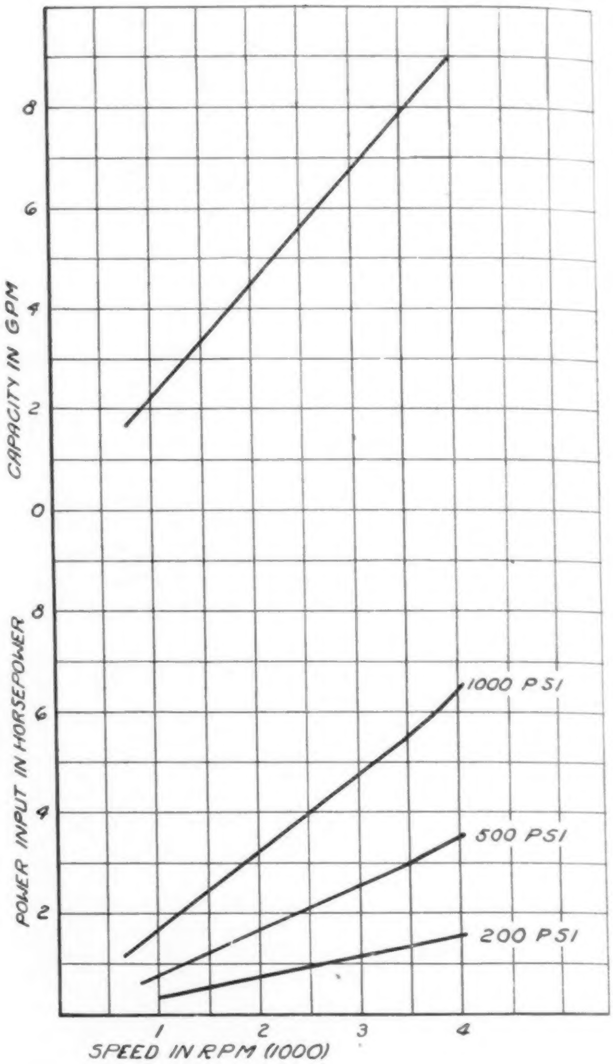
The electric generator is considerably larger for a given output than a hydraulic pump. The space allocated to the generator has been limited to a cylinder 6½ in. in diameter and 15 in. long for several years. A 750-w generator (15 v, 50 amp) was standard during the 1930's in practically all commercial and military applications. This generator is shown in Fig. 4A. During the 1940's a program of improvement was started; and one of the resulting generators is shown in Fig. 4B. The latter occupies the same space and is designed for the same speed as the low-output generator, but it has a rating of 6000 w (30 v, 200 amp). The improvement in the weight-power ratio of generators which resulted from this development is shown by Fig. 5. The generators shown in Figs. 4A and 4B were designed for low speed. High-speed generators are only now being placed in production, since there was a considerable time delay while engine manufacturers inserted a higher ratio between the crankshaft and the generator drive pad.

The trend in efficiency of generators with increase in rating is shown by Fig. 6.

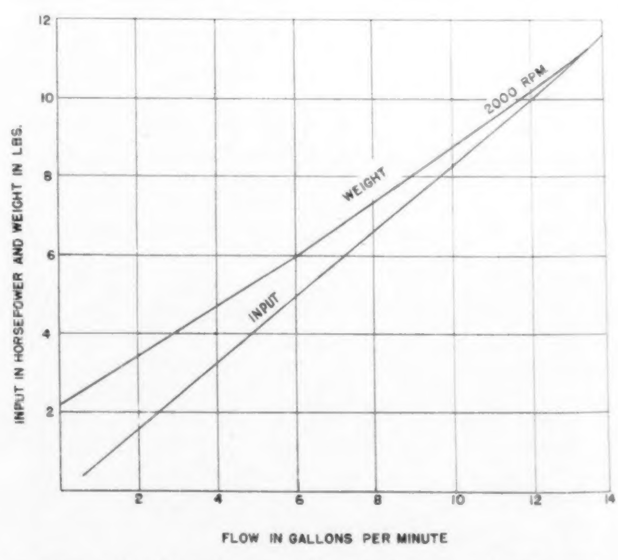
An important difference between hydraulic and electric generators is their speed-load characteristics, as shown in Fig. 7. A hydraulic generator will deliver useful output at its rated pressure at any speed. The flow of fluid will be decreased in proportion to speed, and the output will also decrease in the same proportion. An electric generator



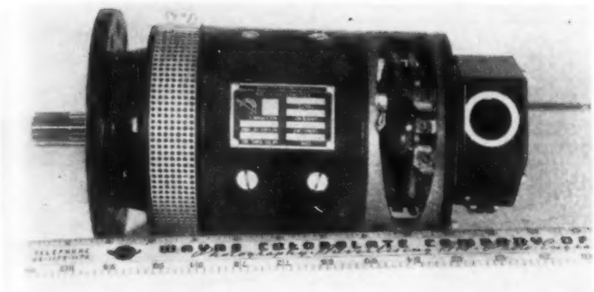
■ Fig. 1 – A typical hydraulic pump



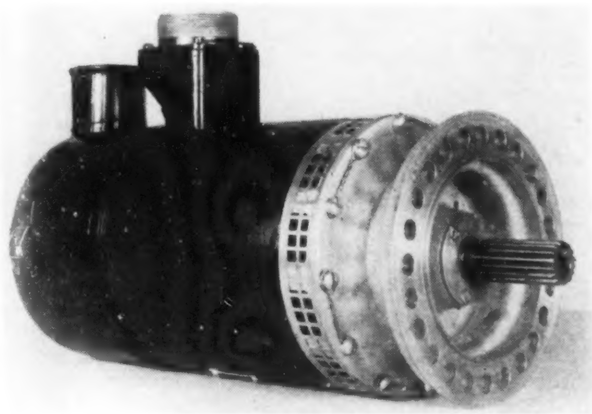
■ Fig. 2 – Performance curves for the pump shown in Fig. 1



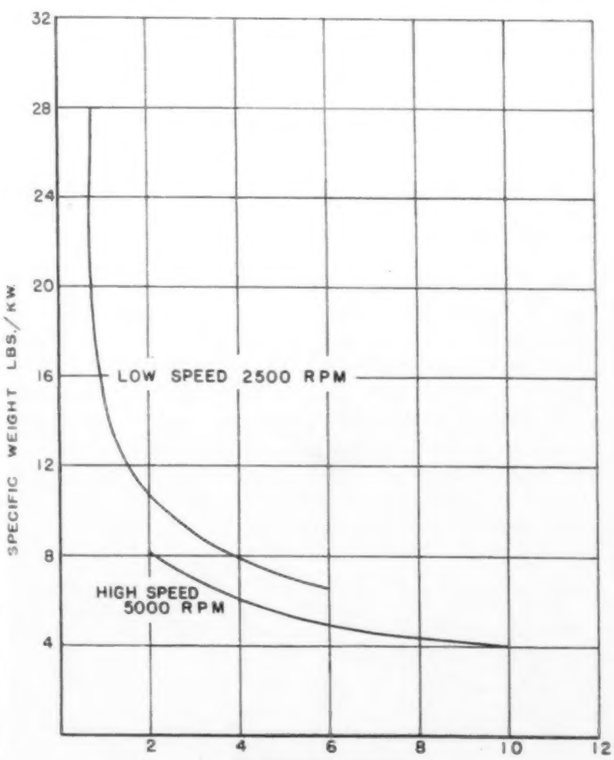
■ Fig. 3 – Weight and power characteristics of hydraulic pumps at 1000 psi



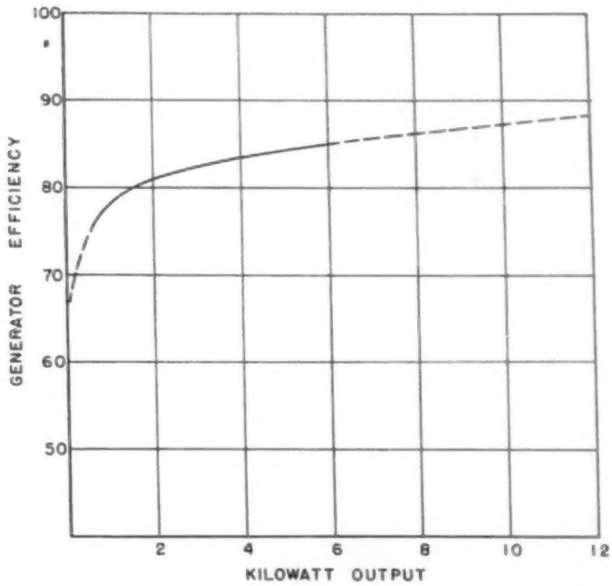
■ Fig. 4A – One of the standard 750-w generators used 1930-1940



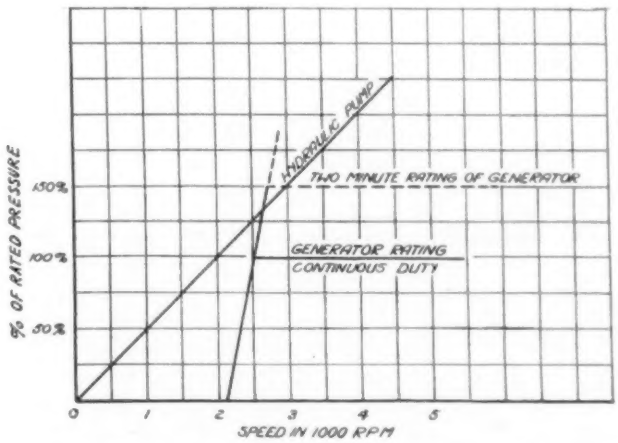
■ Fig. 4B – One of the 6000-w generators developed since 1940. This improved generator occupies the same space and runs at the same speed as the earlier generator shown in Fig. 4A, but it has eight times the power rating



■ Fig. 5 – Weight-power ratio of electric generators



■ Fig. 6 – The trend in generator efficiency with increase in rating

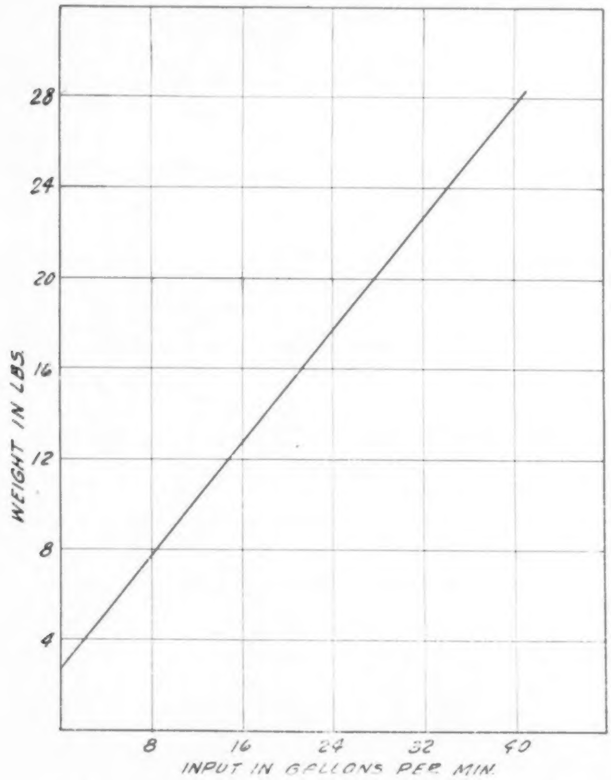


■ Fig. 7 – Speed to develop rated pressure of pumps and generators

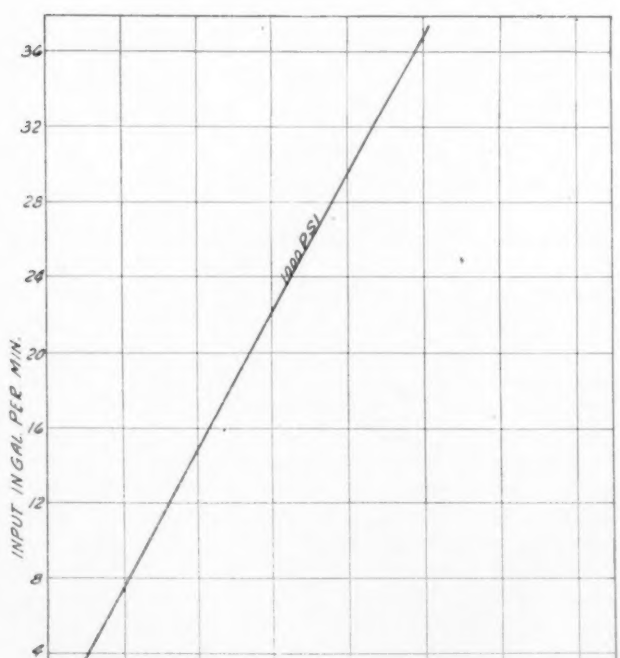
must be operated at a certain minimum speed before it will develop the required voltage. A small increment beyond this minimum speed will enable the generator to deliver full rated load. Any increase beyond the rated minimum speed will permit the obtaining of higher outputs, but only for a short time because the generator would be destroyed by thermal losses within itself.

The device which uses accessory power to drive an accessory may be either a rotary or an axial type. The hydraulic accessory is particularly suitable for the axial type, which may be described as a cylinder and piston. A rotary hydraulic motor is usually a vane- or gear-type pump operated by hydraulic pressure to create rotary motion. Insufficient data are available to permit the plotting of an accurate graph which shows the relation of power to weight and efficiency for hydraulic motors. An estimate of weight is shown by Fig. 8, and of efficiency by Fig. 9. The relation of cylinder weight to operating pressure is shown by Fig. 10.

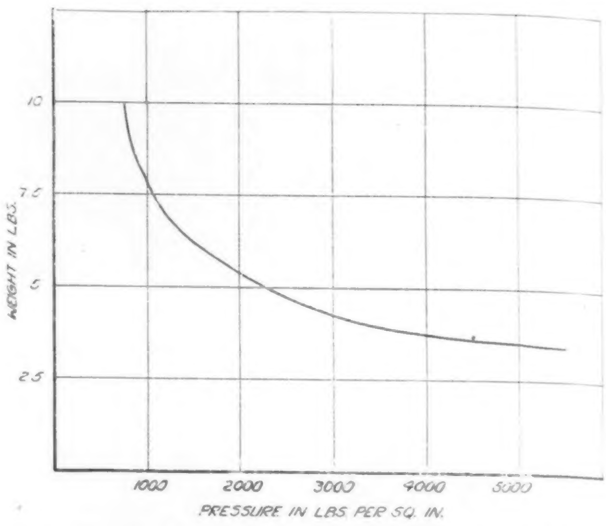
In the case of electric devices, the rotary type is a conventional motor. The solenoid represents the axial type



■ Fig. 8 – Hydraulic motor weight – estimated from pump data

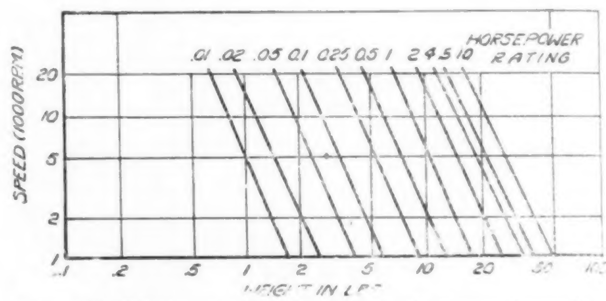


■ Fig. 9 – Performance curve of hydraulic motors at 1000 psi



■ Fig. 10 – Weight of steel cylinder to develop 50,000 ft-lb of work using the best ratio of cylinder diameter to wall thickness

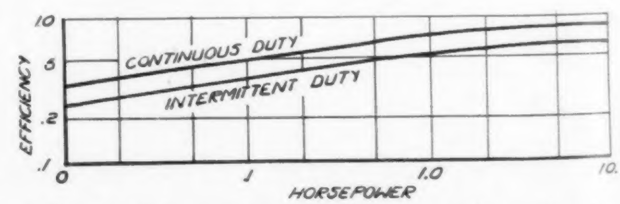
and is not as effective as a hydraulic cylinder since it is limited to very short movements. Most applications for solenoids are restricted to a maximum of ½-in. travel, whereas a hydraulic cylinder can move through several inches. The relation of motor weight to power and speed is shown by Fig. 11. As this curve indicates, the develop-



■ Fig. 11 – Maximum values for weight of d-c continuous-duty electric motors for aircraft

ment of higher operating speeds offers the greatest promise for reducing the fixed weight of rotary-type motors. At the present time, 8000 rpm is a practical limit for continuous-duty motors, and 12,000 rpm for intermittent-duty motors. Manufacturers who use care in finishing and assembling commutators have been able to produce successfully motors to operate at higher speeds. This requires greater precision and skill than most manufacturers have used in their commercial practice.

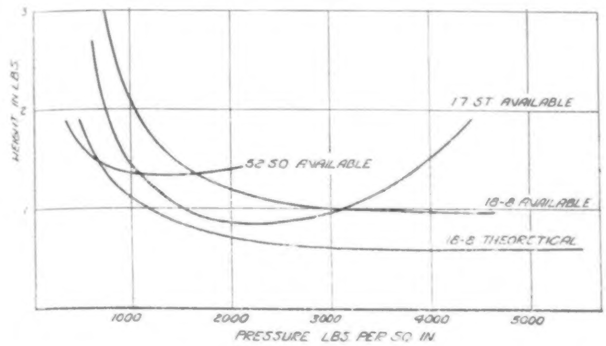
The efficiency of rotary-type motors is shown by Fig. 12.



■ Fig. 12 – Minimum values of efficiency of electric motors for aircraft

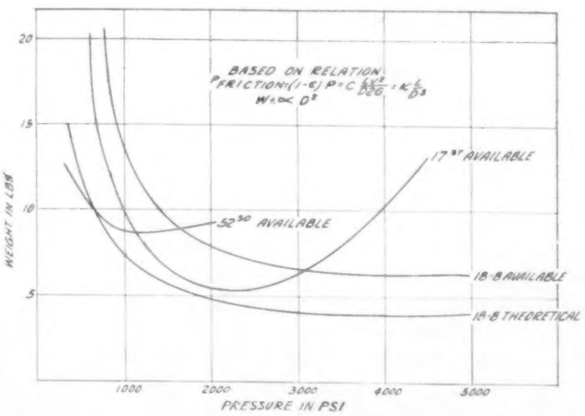
It is the function of the transmission line to interconnect the accessory generator and motor. This line must be efficient, light in weight, easy to install, and simple to maintain.

Hydraulic engineers have been evaluating the weight of this transmission line as that of a 10-ft tubing with oil which has a transmission efficiency of 90%. This relation is shown by Fig. 13A. It seems impractical to consider



■ Fig. 13A – Weight of 10 ft of tubing and oil to transmit 10 hp at 90% efficiency

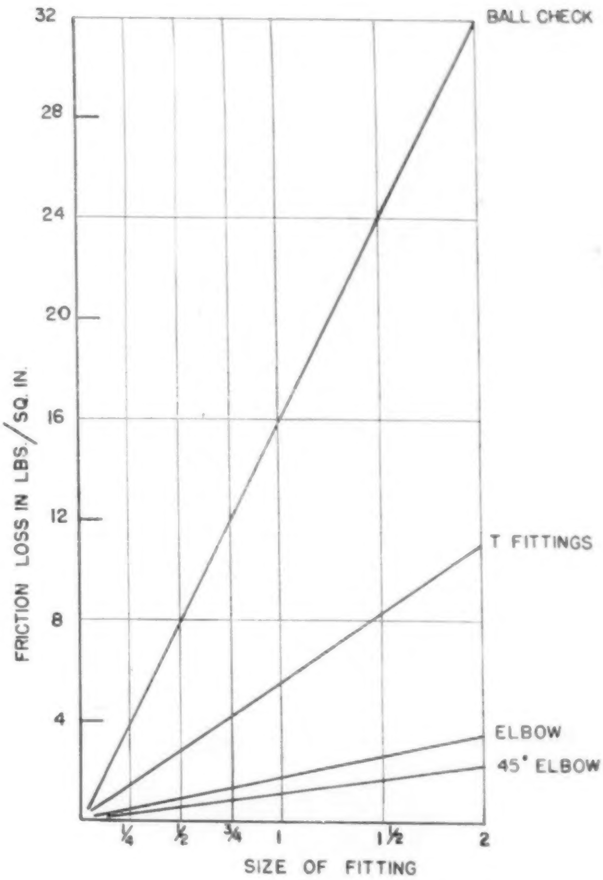
only a 10-ft length because most airplanes which require any amount of accessory power are going to be so large that a 50-ft length is far more reasonable. In addition, a frictional loss of 1% per ft of line is too high to be practicable. On this assumption, a 50-ft line would have an efficiency of 59%. Fig. 13B has been prepared to show the relation for transmitting the same horsepower through a 50-ft line at a 97% efficiency.



■ Fig. 13B – Weight of 50 ft of tubing and oil to transmit 10 hp at 97% efficiency

A second effect which must be considered is that of bends and fittings on hydraulic losses. This is usually evaluated as an increase in the length of the line, and factors therefore are shown by Fig. 14. It is apparent that actual length is not the effective length, as it must be corrected for the fittings and bends which are installed.

With regard to electric lines, it is common practice to allow a 3% loss in the line for continuous-duty applications, and a 6% loss for intermittent duty without regard to length. The problem of the electrical engineer is considerably simplified when the airplane is all metal because

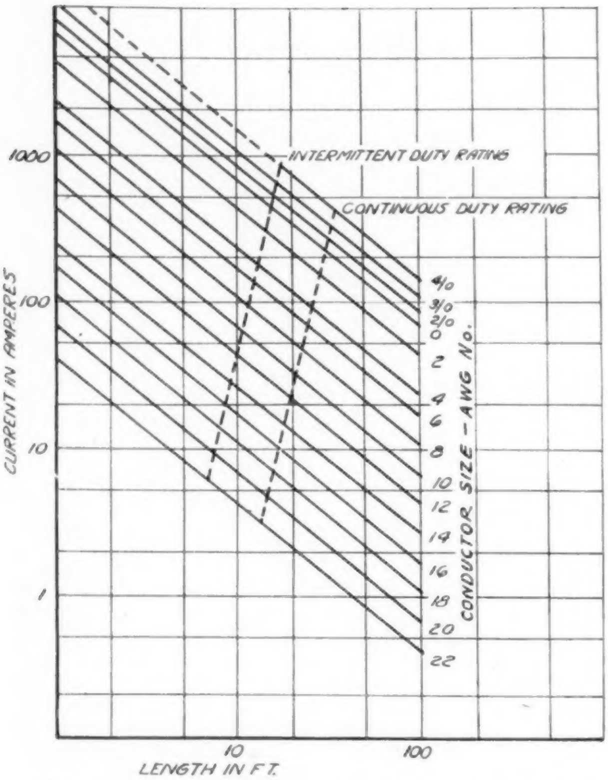


■ Fig. 14 – Friction loss due to bends and fittings

a ground return can utilize the aircraft structure and save the weight of a second conductor. This means that for the same 50-ft line discussed in Fig. 13B, a 25-ft length of electric line could be used. However, the comparison will be made on a 50-ft basis.

The frictional loss in the electric transmission line is the loss due to resistance of the conductor. This will appear as a loss in voltage. According to Ohm's law, this loss will be $E = IR$. Where E is the line loss in volts, I is the current in amperes, and R the resistance in ohms. The resistance of the conductor will be $R = KL/A$. Where K is the coefficient of resistivity of the conductor, L is the length, and A the cross-sectional area. In order to keep the resistance within limits, the electrical manufacturer must adjust the cross-sectional area, A, to correspond to length. The relation between current and length of conductors conforming to American Wire Gage Standards is shown in Fig. 15 for continuous duty. The power transmitted in the lines will be $P = EI$.

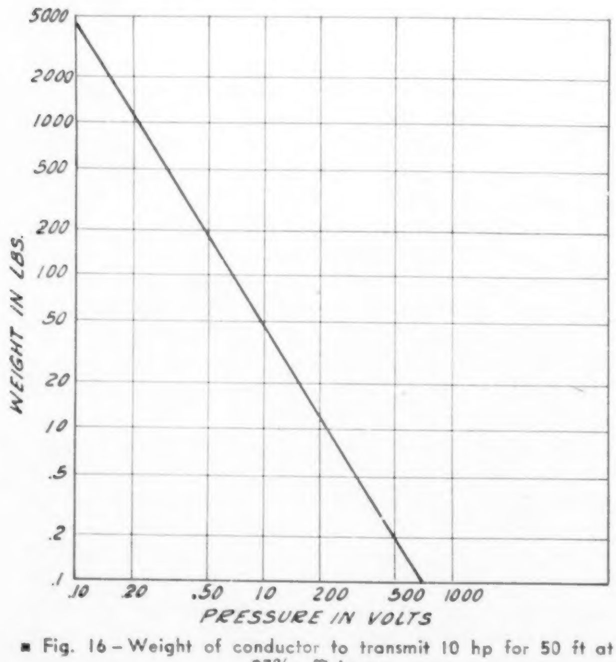
This relation shows the importance of using a higher voltage. The weight of wiring will be directly proportional to the current, and this current may be decreased as voltage is increased. This is the basic reason for using a 24-30-v system in aircraft instead of the 6-8-v system used in automobiles. The electrical engineer has a considerable region for expansion with regard to voltage. A 110-v system represents the next logical step in so far as voltage increase is concerned, and in the case of very large aircraft,



■ Fig. 15 – Relation between current and length of conductors for 3% line drop – 30-v system

an electrical system having a voltage of 440 v is not impracticable.

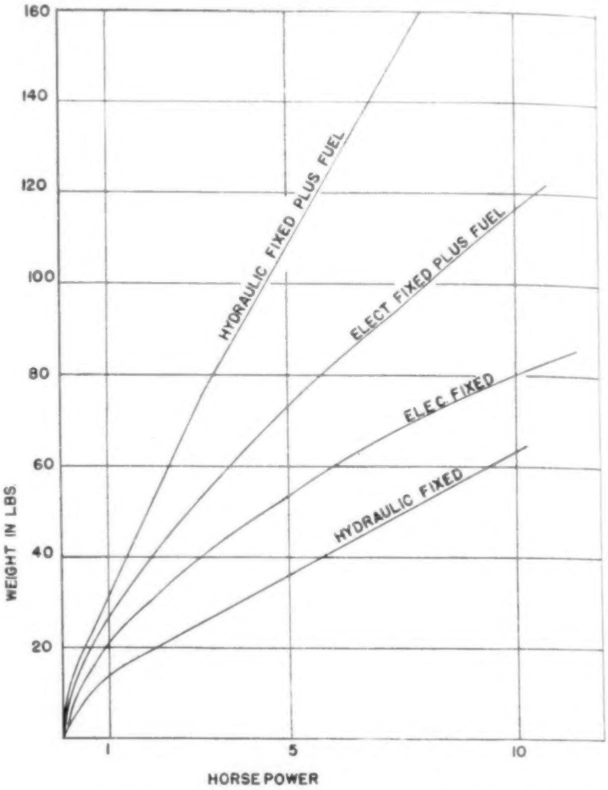
A similar comparison of transmission line weights for a 10-hp load at a distance of 50 ft with an efficiency of 97% is shown by Fig. 16. The relation of this weight to voltage indicates the potential improvement in store for electrically operated equipment.



■ Fig. 16 – Weight of conductor to transmit 10 hp for 50 ft at 97% efficiency

To summarize the comparison between hydraulic and electric power, the relation of fixed plus fuel weight to power required for generator, transmission line, and motor is shown by Fig. 17.

This curve indicates that the fixed weight of electrical equipment for these loads is a handicap. This is due to



■ Fig. 17 – Relation of weight to power required for hydraulic and electrical systems

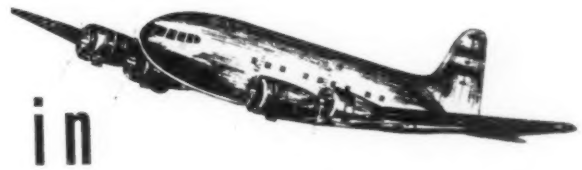
the fact that only a single generator has been provided for the required load. In case this generator capacity is already available, the charge for fixed weight decreases considerably. For example, a 1-hp load will require a 1-kw generator, which in turn will weigh approximately 16 lb. If the additional 1 kw which is needed can be added in a generator already installed which has a rating of approximately 12 kw, the additional rating costs only 4 lb. Again, if the 12-kw generator is required in any event for loads which occur at another period of operation, the additional 1 kw is available at no charge in weight.

This introduces a potential accessory system of a single type which has many advantages. Fig. 18 shows the accessory section of a radial engine as it could be reworked to utilize only one type of accessory power. Since an electrical engineer prepared this article the one type of power considered is an electric generator. This generator would occupy a space approximately 15 in. in diameter and 15 in. long, and would weigh probably 100 lb. Since the electrical industry is now producing generators having a rating of more than 30 kw in a space 6⅜ in. in diameter, 14 in. long, and with a weight of 50 lb, it is quite likely that 100 kw can be taken from the larger space.

A single accessory system will have certain important advantages as follows:

turn to page 247

CABIN SUPERCHARGING in Scheduled AIRLINE OPERATION



by R. L. ELLINGER

Chief Engineer, Transcontinental & Western Air, Inc.

SUCCESS in supercharging airplane cabins for high-altitude flying, Mr. Ellinger explains, came only by solving hundreds of small problems in areas beyond the ken of aeronautical engineering.

The system described by Mr. Ellinger is provided with two engine-driven superchargers, which bring fresh air in from the leading edges of the wings. An alternate system brings in air caught by scoops located on the top of the fuselage. Each of the superchargers has a control unit located inside the cabin.

★ ★ ★

THE AUTHOR: R. L. ELLINGER'S entry into the aircraft industry began in 1929 as a department foreman with the Bach Aircraft Co., Venice, Calif. The following year he went with the Aero Corp. of Calif., a predecessor of TWA, and has been with this organization since that time. For seven years prior to entering the Engineering Department of TWA in 1939, he represented his company's interests at most of the major factories in the United States in the procurement of new aircraft. During that period, and up to the present, he has collaborated in the preparation of specifications which have resulted in the production of many of our leading passenger transport aircraft. Mr. Ellinger, who is now TWA's chief engineer, served with the Marine Corps in France during World War I.

DURING the earlier period of air transport operation, a large portion of the travel was limited to flight under favorable weather conditions only, due, in part, to the lack of suitable radio and other navigational aids, such as blind flying instrumentation and accurate weather forecasting. Even if these facilities had been available, there were few, if any, of the aircraft of that period which had sufficient performance to carry them through or over bad weather.

During this period, the most reliable navigational aids were the railroad tracks; and these, supplemented by the ability of most of the pilots of that era to "fly by the seat of their pants," contributed in no small measure to whatever success was attained in adverse weather operation.

It was very early apparent that air transportation, if it was to be accepted by the public at large, must be safe – it must operate with regularity and it must provide comfort. To achieve this end, operation in that unexplored region above the storms seemed to hold the most promise.

At that time, it was generally considered that most of the bad weather could be topped at 15,000 to 18,000 ft. Consequently, airplanes were developed and introduced which had been designed with that objective in view and which, it was thought, might be suitable for the purpose.

Two of the most popular of these were the Douglas DC-1 and the DC-2, which were designed to specifications prepared by TWA. They had a ceiling of well over 20,000 ft and cruised nicely at 16,000 to 18,000 ft.

These aircraft were, as you all know, a very definite step in advance of their predecessors; however, after a very short period of operation with them, it was apparent that there had been only a slight conception of the numerous problems involved in over-weather operation. It was true that a lot of the adverse weather could be "topped" with these planes – it was also found that a lot more of that kind of weather extended still higher.

Other findings were that the bad weather encountered was of the ice-forming type, more often than had formerly been the case, and also that flight through, and over, long stretches of this weather necessitated the development of accurate blind flying instruments by the industry as well as blind flying procedures and technique by the pilots.

But still more important was the lack of oxygen at the higher altitudes for the crew and the passengers alike. The effect of "oxygen want," or anoxia, as it has come to be called, upon human reactions is rather insidious, as is the effect of intoxicants and, to an extent, is somewhat similar.

It usually affects different persons in an entirely dissimilar manner and at various altitudes.

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

The lack of warning of oxygen want makes it important that steps be taken to prevent the possibility of its inadvertent occurrence.

As a result of our early experience with the effect of anoxia upon pilots and passengers, oxygen was made available to the pilots to be used at their discretion. Later, however, this arrangement was revised by most airlines to provide at least some means for supplying oxygen to such passengers as might require it, as well as to the pilots. These general provisions are still in effect on our domestic aircraft for use under such conditions as might require them to operate above 10,000 to 12,000 ft.

A number of other proposals have been considered for use during high-altitude flight, such as oxygenizing a portion of recirculated cabin air, and pressurizing the individual by means of a rubber suit. These were found to be inadequate from the standpoint of efficiency and comfort, and the pressurized cabin was finally considered to be the most practicable arrangement.

High-Altitude Tests

It was necessary, however, to determine more exactly the altitudes to which we must go to clear the worst weather and also to determine the practicability of operating at those altitudes, and to that end extensive tests were conducted for TWA.

It was found that a large percentage of the "bad weather" could be flown over or around at altitudes in the vicinity of 20,000 ft, although solid cloud formations had been flown through which were 22,000 ft thick, extending to an altitude of 30,000 ft; and still other overcast conditions had been encountered at 36,000 ft and above, which covered approximately one-half of North America.

In these clouds, the so called "precipitation static" becomes increasingly worse with altitude, and at 30,000 ft, radio reception was found to be faulty.

Temperatures as low as —70 F were also in evidence at 36,000 ft. Incidentally, temperatures considerably below this range have given the aircraft manufacturers no small concern in developments projected since then.

The air was normally very smooth at altitudes of 16,000 ft and above, although winds estimated to have had velocities of approximately 150 mph were encountered at altitudes around 30,000 ft.

The use of oxygen, carried in both the liquid and gaseous form, was found to be adequate to maintain normal physical efficiency, with no apparent indication of anoxia, up to an altitude of 30,000 ft, for such effort as the pilot and observers were required to exert. At this altitude, however, practically pure oxygen was required, with complete unconsciousness resulting in about 60 sec, if the oxygen supply were cut off. As a matter of fact, the test engineer on one of the exploratory flights almost met death, as a result of pulling loose his oxygen hose at 30,000 ft and not being able to reconnect it before lapsing into unconsciousness. He did not recover until a descent had been made to 19,000 ft, even after the supply had been restored and oxygen forced into his nostrils.

Pains in the joints, or "bends," as they are called, were occasionally experienced above 30,000 ft. This condition was in no way affected by the use, or lack of use, of the oxygen equipment provided.

Speed increases of roughly 25% were attained at 27,000 to 30,000 ft.

The data obtained on these flights were relatively very meager; however, they gave an indication of many possible advantages which might be gained in high-altitude operation; they also indicated that oxygen was not the solution to safe and comfortable passenger transport operation.

As a result of the tests and exploratory flights, the development and production of the Boeing Stratoliners were fostered by TWA, and a number of this type of plane were delivered to TWA and Pan American Airways in the spring of 1940, and scheduled operation on our system was begun in July of the same year. Most of you are perhaps familiar with these aircraft, since the cabin supercharging system employed has been previously described in detail by James B. Cooper of the Boeing Aircraft Co.[1] However, I will describe briefly the fundamentals of the aircraft and the supercharging provisions.

The airplane was a 4-engine craft of all-metal construction, with a maximum gross weight of 45,000 lb, as approved by the Civil Aeronautics Administration. It was powered by Wright Cyclone G-105A engines, equipped with two-speed superchargers. These engines provided 4400 bhp for take-off, and furnished sufficient power to permit cruising at altitudes of 21,000 to 23,000 ft.

The normal crew consisted of two pilots and one flight engineer, with two hostesses to care for the passengers.

A maximum of 33 passengers could be carried in the daytime and 25 at night, 16 of whom could have sleeping berths, with the remaining 9 occupying reclining chairs.

The cigar-shaped fuselage was circular in section throughout, which simplified the structural design from the standpoint of withstanding the stresses imposed by pressurization. The nose of the fuselage also carried out the streamline effect, there being no offset at the cockpit windshield as in many other planes. The fuselage was pressurized in its entirety from the nose rearward to a location aft between the main cabin door and the leading edge of the horizontal stabilizer. At this point, a pressure bulkhead separated the pressurized portion of the fuselage from the tail section to the rear.

Sealing

It was necessary to reduce the amount of air leakage to a minimum, of course, and special attention was given to all points where leakage might be possible.

The skin seams of the fuselage were sealed with strips of tape between each joint.

All exterior doors opened inward, to simplify both the structural and the sealing problems involved. These doors were provided with a seal utilizing an edge of a sheet metal strip pressing against a hollow rubber tube, the latter being held in place by a sheet metal channel, or retainer. The relatively small area of surface presented by the edge of this metal seal strip required comparatively little pressure to prevent leakage, and since the doors opened inward, the greater the pressure, the tighter the doors were sealed.

Cable controls, passing out of the pressurized area, were sealed very effectively by means of a simple rubber or neoprene seal, through which each cable passed.

Plumbing or tubing, passing into or out of the pressurized area, was provided with special connector blocks and fittings where they passed through the fuselage skin.

Electrical lines passed through junction boxes and were attached to connector blocks at the skin line.

Wash basins were equipped with spring-loaded drain valves so that no pressure would be lost except when the valve was being held open to permit the water to drain out.

The instrument vacuum system was provided with a special pressure-reducing valve.

The fuselage structure was not appreciably heavier than would have been required for a nonpressurized cabin, except for the reinforcing required around door and window openings. The cockpit windshield glass, however, was ⅝-in. safety glass.

Since a large portion of the forward end of the fuselage was cut away to permit installing the windshield, it was necessary to increase the strength of the small remaining strips between the windshield panels by the use of stainless steel. This particular type of material was used to avoid any unfavorable magnetic reaction on the compass, as would have been the case with ordinary steels.

The cabin supercharging equipment consisted of two engine-driven centrifugal blowers, one of these being con-

[1] See SAE Transactions, Vol. 48, June, 1941, pp. 240-248; "Altitude Conditioning of Aircraft Cabins," by James B. Cooper.

nected, through a reduction gear, to the generator drive of each inboard engine. During take-off, the impellers of these superchargers operated at over 27,000 rpm. At cruising speeds, however, this was reduced to between 17,000 and 21,000 rpm, depending upon the engine speed chosen. Each blower had a rated output of 225 cu ft of air per min at an absolute pressure of over 9 psi, an air temperature of 70 F, and an engine speed of 1500 rpm, which was equivalent to supercharged operation at approximately 19,000 ft. This output furnished roughly 12 cu ft of air per min per person under maximum passenger conditions.

The air entered the superchargers through ducts which picked up fresh air through openings in the leading edge of the wings. These ducts were provided with screens to keep out foreign objects and with suction relief valves which would open in the event the normal opening became clogged with snow or ice. These screens and the suction relief valves did a very good job and served a worthwhile purpose – as we found out when the screens froze up early in our operation, thereby shutting off the airflow to the cabin very effectively. We found they had been installed on the wrong side of the suction relief valve.

The air was discharged from the superchargers into ducts which led the air through steam radiators, into the supercharger control units in the fuselage, and from there through other ducts to various parts of the cabin. These ducts were provided with pressure relief, or surge, valves, which were designed to relieve the duct pressure if it became too great, and which were controlled by pressure and suction lines from the supercharger control.

It has been generally considered by house ventilating engineers that 5 cu ft of air per min per person is sufficient for normal ventilating purposes. This figure cannot be subscribed to for aircraft work; however, it was found that the output of one of the two superchargers in these planes was adequate for the purpose under most conditions.

As an alternate to the ventilating airflow obtained from the two superchargers, it was possible to obtain air directly from an outside airscoop on top of the fuselage, which introduced the air so obtained into the regular ventilating system within the plane. This provision was made for use when supercharging was unnecessary, or perhaps undesirable, because of uncomfortably high air temperatures. Aside from this alternate source of air, there was no provision for cooling cabin air or air from the superchargers.

The cabin supercharging control was composed, briefly, of one control unit for each supercharger. These were located within the cabin on each side of the fuselage. Each of these was, in turn, composed of two sections, one section of which controlled the airflow into the cabin by means of balanced valves. These balanced valves were controlled by predetermined pressure and venturi suction forces, interconnected to the blower relief, or surge, valve, located in the duct close to the supercharger. Each inlet valve was also manually controllable from the cockpit. The other section of the control unit regulated the cabin pressure by restricting the outflow of the cabin air. This, too, was done by means of a balanced valve. In this case, however, the activating force was obtained by means of a sylphon bellows during that portion of the pressure control range from 8000 to 14,700 ft, and by a spring set to maintain a constant differential pressure of 2.5 psi above that point. Both of the main control units were so connected that if one supercharger were to fail the other would carry the load.

For heating the cabin, a steam system was provided, which had two steam boilers located in the exhaust tail pipe of each inboard engine. The air from the cabin superchargers passed through the radiators, or heat exchangers, of this system before entering the cabin.

The first of these supercharged aircraft was tested in the winter and spring of the year and delivered to TWA in May of 1940. I point out the time of the year the tests were conducted because of its significance upon later events.

Inasmuch as these aircraft were an entirely new type of equipment they came in for a series of functional tests and a lot of critical examination by our various operating departments, before any attempt was made to put them in operation. As a matter of fact, it was even necessary completely to revise the galley first. Those of you who may be familiar with the history of airline galley requirements will know this was no reflection on the builder.

Flight engineers were used by TWA for the first time on this equipment. Even in the early days during operation of the 4-engine Fokker F-32, it had not been thought necessary to use flight engineers. However, the cabin supercharging and other complications such as steam systems, hydraulic systems, de-icing equipment, engine cowl flaps, retractable landing gear, and wing flaps had added considerably to the duties of the pilots and rendered the use of a third crew member in the cockpit a necessity.

The training of the first group of flight engineers was not a small problem, especially as concerned the cabin supercharging operation, inasmuch as it was entirely new and untried. This training was accomplished, however, by sending some of the men to the factory, before the delivery of the equipment, to study the manufacturing phase of the development, and by setting up, several months in advance, special courses for their education in all branches of aircraft operation with which they might come in contact. They were even required to obtain sufficient knowledge on the side to enable them to pitch in and do an excellent job of substituting for the hostess as the occasion demanded. The thoroughness and soundness of this training was reflected in the high degree of success attained from the beginning of the operation.

The pre-schedule flights and tests were all very favorable in general, until just a few weeks before the date set to start operations; then, the first taste of summer weather arrived, along with a very marked temperature inversion aloft. We soon discovered that the amount of heat generated by compressing the cabin air was much more than could be dissipated through the air ducts before entering the cabin. This demonstrated conclusively that tests to determine the cooling efficiency of a ventilating system should not be made in the wintertime. Fortunately, our engineering department had had a premonition of just such an event, in spite of the favorable reports during the tests and simulated scheduled flights, and they had completed the development of supercharger intercoolers. These were fabricated and installed in all planes within a period of three weeks, permitting our scheduled operation to start.

■ Maintenance and Testing

Prior to the inception of our operation with the supercharging, we were somewhat concerned as to the maintenance attention which might be found necessary, and the overhaul procedures that might be found advisable to obtain the maximum efficiency. Above all, we were concerned with the testing equipment and technique. As I have said before, it was all untried and unproved. There

had been no precedents established upon any part of the supercharging components, with the possible exception of the blowers themselves. These were quite similar in principle to the blowers which had been and are still used successfully to supercharge aircraft engines, and strangely enough, some of our most serious difficulties involved failures of these units.

There were at that time no elaborate high-altitude pressure chambers, of which there are now a number scattered throughout the country, in which the cabin pressure control units might be tested periodically, and after overhaul, to see if they were operating properly. As a matter of fact, to my knowledge, the only altitude chamber of any kind available at that time was the 50-gal drum which had been used in the original development of the control units.

Since we had but a few of the controls to service and it appeared likely that it would be a long time before there would be more, it was obviously not good economy to provide elaborate testing provisions for these few units. The first few which required overhauling were returned to the manufacturer for that purpose. We soon found it impossible, however, to keep sufficient spare assemblies on hand to permit this. Consequently, we did the next best thing, and which we believe is, up to a certain point, quite practical; we used the airplane itself for the testing laboratory – and obtained excellent results. As a matter of fact, there was but very little to go wrong with the controls which might require a pressure chamber for testing. The sylphon bellows and the few springs in the control units could be readily checked in the instrument shop. If they were within their tolerances, and if the clearances and tolerances of all of the other parts were within limits, and also if the various valves moved freely, the operation of the controls was pretty sure to be satisfactory.

The testing of the superchargers themselves and the pressure sealing of the cabins came to pretty much the same end. By the time test equipment considered suitable for routine maintenance and overhaul processes had been devised, we had encountered such excellent results by simply using the aircraft and its instrumentation as the testing "laboratory" that the special shop equipment was never completed. As a routine procedure, these tests were coordinated with the periods required for engine changes, and supercharging testing was accomplished on the same test flight as that required for the engines.

Testing of the cabin supercharging consisted simply of observing the airflow from each supercharger, under controlled conditions, as indicated on a vacuum gage on the flight engineer's panel. The cabin pressure and rate of pressure change was also indicated.

The rate of leakage was readily determined by shutting off both superchargers, with the cabin pressurized, and then noting the rate of pressure change, as indicated on the flight engineer's vertical speed indicator. The rate of leakage, established as a maximum, was that which would be equivalent to a rate of ascent of 1000 fpm.

Cabin Sealing

The sealing of the fuselage skin laps and joints was very satisfactory and no difficulty was experienced from that source during our period of operation, even though the tape which was used to seal the laps was soluble in oil and gasoline and eventually showed a disposition to deteriorate along the edges of the joints.

The sealing of the control cables, where they passed through the fuselage shell, was also very satisfactory.

The door seals, however, were not as maintenance-free as the other portions of the airplane due to the fact that they were made of spliced sections of rubber tubing, and the structure for retaining the seals, as well as the mating seal flange, were of very light material which was easily damaged. However, except for the lightness of the material used to secure the door seals, the design was very satisfactory and much more preferable to using cement.

These door seals were all readily accessible to visual inspection and it was possible in most instances to determine whether they were satisfactory by simple observation.

Supercharger Problems

There were a number of relatively minor difficulties experienced with the supercharger control units which I will attempt to cover; however, these seldom resulted in any appreciable discomfort or inconvenience to the passengers due to the fact that the capacity of one supercharger was sufficient to handle all normal requirements, in the event of the malfunctioning of the other unit.

The main body of the control units was composed of cast magnesium. Two principal sources of difficulty with these units was occasioned by warpage of the case assemblies and with corrosion between the stems of the balanced valves and the guides, the latter of which were bronze bushings pressed into bosses on the magnesium casting. This corrosion naturally caused sluggish operation at best, and since the valves were rather accurately balanced to permit operation by the relatively light forces imposed by the airflow and venturi suction, it naturally required but very little force to restrict their movement.

Some sluggishness was also caused in these valves by a gummy deposit, lint, and dirt found adhering to their stems, which were deposited there by the airflow.

No solution for the elimination of these particular problems was found except cleaning and maintenance.

The balanced valves required a predetermined rate of flow past and around the pistons to provide satisfactory operation. This necessitated the maintenance of closely controlled tolerances between the pistons and the side walls. We found it was impracticable, due to warping of both the piston and the magnesium castings, to maintain these tolerances, and so increased them slightly, with but a moderate improvement in operation.

The characteristics of the pressure relief, or surge, valves in the supercharger outlet ducts, and the forces controlling these valves were of such a nature that in operation around 20,000 to 21,000 ft and above, any slight restriction or irregular operation tended to open the surge valve and keep it open, thus throwing one supercharger out of use and reducing the volume of ventilating air going to the cabin. Fortunately, there were very few, if any, instances of this happening to both superchargers at the same time. It did happen too frequently, however, with one supercharger or the other, and it was considered expedient, eventually, to forego the attempt to perfect fully automatic airflow regulation with this particular type of control, and we finally revised it to provide completely manual control.

However, the automatic feature of the pressure regulator, or outlet valve, was not changed, and this portion of the control functioned with a reasonable degree of satisfactoriness during the remainder of our operation.

An unanticipated result of this manual operation of the flow control resulted in almost immediate complaints from

the passengers, hostesses, and the flight crew, of excessive dryness of the skin and burning of the eyes. The flight engineers, in their endeavor to provide the maximum in comfortable ventilation, mistakenly operated the superchargers at a much higher flow rate than normal, and the output at normal altitudes of around 18,000 ft was so much higher than was actually needed for the average passenger load that the air in the cabin was dehydrated to the point that it was uncomfortably dry. This was easily corrected by a word of instruction to the flight engineers and the establishment of a normal maximum rate of flow.

As I have mentioned before, some of our most serious and costly difficulties were encountered in the superchargers themselves. These failures involved the main impeller bearings and due to the nature of the parts and the installation, it was impossible to foretell just how long the bearings might last. The manufacturer recommended an overhaul period of 250 hr for the superchargers at the beginning of the operation, and to my knowledge nothing was ever found particularly bad at any overhaul periods, even after they had been extended to correspond with engine overhaul periods of 650 hr. Nevertheless, we encountered a number of bearing failures. Some bearings failed within 200 hr, others operated for 1200 to 1500 hr.

The failure of a single bearing on one of these units might appear to be a rather inconsequential item; the bearing itself cost but a few dollars to replace and the loss of the output of one supercharger was of no immediate concern since the other supercharger would carry the load satisfactorily. But when a bearing did go out, it inevitably threw the impeller out of alignment, thereby damaging the impeller, the diffuser, and the housing. Also, particles of the damaged bearing and other parts got into the gears and other moving parts, with the result that a bearing failure usually meant a completely ruined supercharger – and the cost of those superchargers was no small item.

Due to the fact that these superchargers were driven directly from the engines, it was not possible, in the event of failure, simply to throw out a clutch to stop or disconnect the supercharger and then continue the flight at whatever altitude might be practicable; it was necessary to feather the propeller on the engine involved so as to minimize the damage until arrival at the next stop and then disconnect the supercharger drive, whereupon the flight might be continued at normal altitudes until a replacement was available. This was the most serious of the difficulties encountered with the supercharging system, as it very obviously defeated one of the basic advantages of the 4-engine operation by requiring that one of the engines be cut off for a reason which, in itself, should have been of no particular concern. It should not be necessary to hamper scheduled operation because of the failure of equipment of this type; it should be possible to disconnect or stop the cabin supercharger in the event of difficulty.

On the whole, the difficulties encountered with cabin supercharging, and the cabin supercharging equipment, were really inconsequential and certainly presented fewer obstacles to successful operation than had been anticipated.

■ Other Problems

There were other problems involved which were not directly connected with cabin supercharging, but which came to our attention during this operation and which must be taken into consideration for high-altitude, low-temperature flight. The problems connected with low temperatures and a reduction of air density are both worthy of mention, particularly as concerns temperatures, since previous tests indicated that temperatures as low as −40 to −60 F were undoubtedly encountered and for comparatively long periods of time, although we have no record of the actual temperatures due to the range limitation of the standard airplane temperature-measuring equipment.

■ Hydraulic Power Boost

The hydraulic power boosts for the main control systems were somewhat of an innovation and came in for quite a little discussion; some pilots liked them, some were just the opposite, and some were noncommittal. They were, nevertheless, accepted by all as necessary, and they functioned with an acceptable degree of satisfaction until the winter weather started in, when we suddenly found ourselves confronted with a formidable list of confusing reports of erratic elevator control operation. The importance of smooth and effective action of the elevators during take-off and landing naturally subject them to the most critical observation by the pilots.

The difficulty resulting in these reports was quickly traced to the effect of low temperatures upon the system.

Part of the trouble was caused by the valve which was used to turn the boost control on and off. This valve, as well as the control boost cylinders, was located in the rear portion of the fuselage outside the heated area. The same type of valve was used on both the rudder and elevator boost. It was a straight bore, plug-type valve, the steel plug of which rotated in an aluminum-alloy casting. The parts were machined to fairly close tolerances and, due to the differences in the coefficients of expansion, they seized after getting down to temperatures near 0 F. By increasing the clearance between the valve body and the housing, this difficulty was readily cleared up.

The remainder of the trouble was due to the temperature effect upon the entire boost control system. Upon encountering temperatures below −10 to −15 F, the viscosity of the oil in both the rudder and elevator boost control systems increased to the extent that it was very difficult to operate them. This condition was finally corrected by providing a circulating oil flow through the power boost system, thereby maintaining the oil at a suitable viscosity.

By the time we had been able to correct these difficulties, it was decided that the proper thing to do would be to eliminate the main cause of the trouble, so we removed the 26-odd pounds of miscellaneous valves, plumbing, and boost cylinders, and operated the planes from then on without the elevator boost control. To accomplish satisfactory control under this condition, it was only necessary to restrict the forward loading of the planes, which happened to be no particular handicap.

Another item of minor importance, which gave us a little difficulty, was the freezing of the wash basin drain lines. I mention this as an item of minor importance because it appears to be, at first glance, something readily corrected by almost any ten-year-old boy. However, at the time, we had no ten-year-old boys and tackled it the hard way.

There were three water drain lines extending through the fuselage, one from the wash basins in the men's room, one from the women's room and one from the galley. To reduce the possibility of the water from these drains collecting on or in the fuselage, they had been extended from the skin line about 7 in. and turned toward the rear.

Some time after our entry into the first winter's operation, we received occasional reports from the hostesses that the wash basins would not drain. Upon finding that this occurred at a time when the outside air temperatures were fairly low, the immediate conclusion was that the outlets had frozen – with that length of tubing sticking out into subzero temperatures, that seemed to be the obvious cause. We thereupon cut them off close to the fuselage, thinking that conduction from the warm region inside would surely keep the lines from freezing. We received no reports for a few weeks and were just congratulating ourselves on the ease with which we had overcome the difficulty when the temperature dropped again, and once more came the reports of the wash basin drains freezing. We were skeptical about the water freezing inside the drain tubes, especially as they were so close to the heated part of the cabin; but when the flight engineers reported that they picked ice out of the outlets with an ice pick after landing, and opened them up again, we took the matter seriously.

We next installed several different means designed to prevent freezing of the drains and to thaw them out. During the next few weeks we had the flight engineers deliberately attempt to freeze up the drains – they even carried extra quantities of water along for that purpose. Not a single drain was frozen; in the meantime, reports came in of frozen drains on the other planes.

Finally, a plane was set aside for test purposes to the end of arriving at the solution quickly. Each of two of the drain outlets on the plane had a different proposed method of keeping it clear of ice and one of them had two such arrangements. One had an electric heater and, in addition, a clam shell over the opening; another had a large diameter outlet installed which extended inward a short distance, the thought being that with the warm cabin air all around it, it would not freeze entirely over; the remaining one was left as originally designed, with the 7-in. extension protruding from the fuselage.

The test flight was made – it lasted an hour or more at an altitude of 20,000 ft and with an outside air temperature of −17 F – during which time gallon after gallon of water was poured through the drains, sometimes fast, sometimes slow, and still the drains functioned normally. We ran out of water and had to come down. After the plane came to rest in front of the hangar, where the temperature was considerably below freezing, we discovered great lumps of ice hanging on the belly of the plane where the water had run out and built up; but not a drain had frozen, not even the one with the original 7-in. extension. Some of them were pretty well choked up, however, and might have eventually frozen closed if we hadn't run out of water and had to come down.

This would appear to have been a rather discouraging and fruitless test – and it was – but "something new had been added" which altered our approach to the problem; that something was the extremely large lumps of ice we found hanging to the outside of the plane. We decided then that we didn't care whether the drains kept clear or not, if the water they discharged was going to build up such large lumps of "go-slow" on the outside of the plane.

A subsequent study of all of our test data indicated that practically all of the tests made by both our engineers and the flight engineers had been inadvertently conducted in the daytime with the sun shining, and even though the outside air temperatures were −17 F and much colder than at such times as the drains had actually been frozen, the heat transmitted around the periphery of the plane, and perhaps as reflected from the clouds below, apparently prevented the drains from freezing during the tests.

So – with the first winter well along – we started all over again, this time with the idea of using a receptacle or tank inside the heated area of the fuselage. The idea of the tank is pretty simple – the drain line connects into the top of the tank and out the bottom, then out through the fuselage. If the drain outlet doesn't freeze the water runs right on through; if it does freeze, the water will run into the tank and stay there until a warmer outside air temperature is reached and the outlet thaws out. I can't say just how successful this system might be, however, because we never got to try it.

■ Cabin Heating

As mentioned before, one of the first problems which confronted us in the operation of this equipment was that of reducing the temperature of the air from the cabin superchargers. With the advent of colder weather this heated air took on an entirely new significance because we found that the steam heating system, which was not used at all during the summer months, could not be made to operate consistently and satisfactorily in the winter months, and we were exceedingly grateful for the heat generated by the cabin superchargers. During the first winter's operation we had no recourse but to get along as best we could with the heating system as designed and to utilize to the fullest extent the supercharger heat.

Inasmuch as we occasionally have extreme summer temperatures at our western terminals, and winter temperatures aloft and at our eastern terminals at the same time of year, it was necessary not only to be able to use the supercharger intercoolers to reduce the air temperature, but also to be able to eliminate them so as to increase the cabin temperature as needed. To this end, the supercharger duct system was provided with bypass valves, so the air could either be routed through the intercoolers or directly into the cabin. By this arrangement, the cabin could be heated comfortably by the superchargers alone, with the outside air temperature at −10 F.

Prior to starting the second winter's operation, this was supplemented by installing a 40,000 Btu internal-combustion heater in each cabin supercharger air duct, in place of the original steam system. This combination, together with the selectivity of the original ventilating system, provided the best aircraft cabin heating and ventilating system we have ever encountered.

■ Moisture Absorption

The opinion is held by some that in supercharged cabins, particularly in a design such as this one, where the warm air from the cabin is exhausted through the belly and where extremely low outside air temperatures are encountered, the moisture from the warm air might condense on the inside of the fuselage skin in comparatively large quantities and perhaps contribute to an increase in weight due to moisture absorption by the insulation, as well as providing a possible source of corrosion.

We had occasion to remove some of the insulation in these planes after they had been in operation about 18 months, and samples of the various types of insulation from all over the aircraft were weighed carefully. There seemed to be little difference in the weight increase of that installed in the cabin as compared with that in the cargo

compartments under the cabin. On the whole, however, the weight of all insulation, which was of the type commonly used in commercial aircraft, increased approximately 35% in the 18-month period of operation. This amounted to some 160 lb per plane or over ½ of 1% of the empty weight of the airplane.

We have checked the weight of the same type of insulation in our DC-3's which are not supercharged, and this has shown an increase of approximately 28% in five years. This difference indicates that perhaps either the cabin supercharging or the operation at comparatively lower outside air temperatures may be factors tending to increase the condensation.

Evaluation of Results

In evaluating the results of our experience with this equipment, not only as concerns moisture condensation in the fuselage but other aspects of the operation as well, such as cabin leakage control, supercharger control operation and supercharger size, it should be pointed out that this system is a full-flow ventilating system, passing and controlling, both as to flow and pressure, a relatively large volume of air upon which minor leaks and small changes or variations would have but little overall effect.

Consequently, the problems involved may be quite dissimilar and require entirely different handling if a different system were used. For instance, if the cabin air should be recirculated, the supercharger capacity might well be much lower. As a result, the cabin would require close leakage control because of the lower capacity; the humidity and resulting condensation might be higher, as a lesser amount of moisture would be discharged overboard; the air filtering and cooling equipment would add new problems; if the cabin were smaller, such as for but one or two persons, the flow and pressure control problem might become more exacting because of the smaller volume involved. Then, too, if higher altitude of operation is contemplated, the cabin pressures will need to be increased, with a resulting increase in the weight of the fuselage structure.

Other items, too, need to be considered for operation at higher altitudes. The fact that the air keeps getting lighter is no less important from the standpoint of the human body than it is for the satisfactory operation of the engines. They, too, must be supercharged still more in order that they can develop the power necessary to keep the plane in flight; and the more power they develop the more heat they must dissipate, and still more of that thin air is required to keep them cool.

In pointing out some of these things and in describing some of the difficulties encountered in our supercharged cabin operation, it might appear that perhaps this development is of questionable merit and its operation impracticable. That, however, is very definitely not the case. It was very highly successful, particularly as concerns the cabin supercharging as such, and we were very agreeably surprised that we had no more difficulty with it than we did.

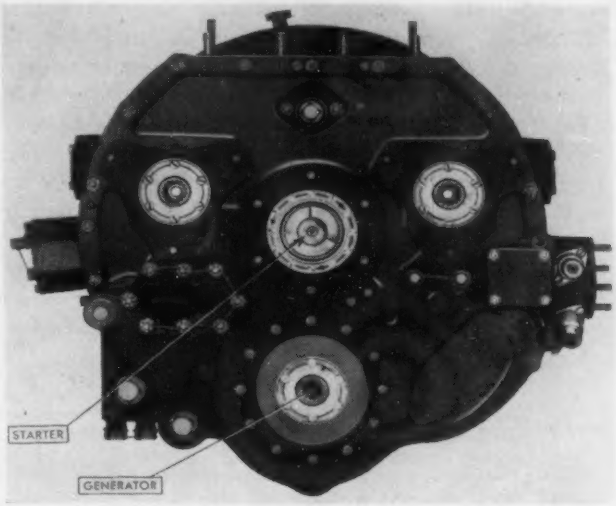
The ability of the Stratoliners to fly over long stretches of adverse weather resulted in these craft being the only planes in the air many times during our operation of them – the unsupercharged planes having been grounded because of their inability to clear the weather with safety and comfort for the passengers at the altitudes which were necessary.

A check of our operating records for one six-month period, from January to June, shows that during that time 22.5% of all Stratoliner flights were conducted at altitudes at or above 10,000 ft, the highest having been at 23,000 ft and which was for but a short period of time. Of these flights, during which cabin supercharging was necessary, 21% were made at 10,000 ft, 16½% were at 11,000 ft, 14% were at 15,000 ft, and 17% were at 17,000 ft, indicating that quite a few were at altitudes which would not have been attainable without pressurized cabins. This is particularly significant when it is remembered that this type of equipment was planned only to permit us to operate at higher altitudes when it was necessary because of passenger comfort or the completion of schedules.

The popularity which these aircraft enjoyed during their entire period of commercial operation speaks eloquently for the success of that type of equipment, especially as far as passenger comfort and convenience are concerned, and there is no question in my mind but that pressurized cabins will be a prime requisite for all passenger-carrying aircraft.

Aircraft Accessory System

continued from page 240



■ Fig. 18 – Accessory section of a radial engine reworked to utilize only one type of accessory power – an electric generator

a. The generator efficiency will be increased, which in turn will decrease the amount of fuel consumed during the aircraft mission.

b. The weight-power ratio will be further reduced.

c. The internal construction of the engine will be greatly simplified.

d. The total weight of the accessory system will be decreased.

This suggestion is submitted to the aircraft industry as a focal point for discussion and comment. The accessory system in modern aircraft has been too often controlled by separate groups of engineers, each believing in his own method, with no control on the complete installation. It may be possible, by the use of a single accessory power, to place the accessory system under one control and obtain a maximum benefit with regard to weight, efficiency, reliability, maintenance, and cost.

SHOT BLASTING TO INCREASE

WHILE great strides have been made in most phases of engineering and metallurgy, it is doubtful that in dynamically loaded parts we are getting more net work from our metals today than was obtainable 25 years ago. The fact that modern airplane engines weigh only about one-half as much per horsepower as the engines of World War I is primarily due to improvements in fuels and increases in engine speed. The speed and performance of airplanes have increased because of the better power-weight ratio of engines and aerodynamic improvements in propellers and airplane structures. New fabrication techniques have made possible many design improvements, better bearing materials are available, lubricants have been improved; but the basic useful strength of our structural materials remains unaltered.

Although no super-strength alloys have been discovered and no such discoveries seem to be imminent, there is much that can be done to increase materially the fatigue strength of many machine parts made from our ordinary structural materials. This fatigue strengthening does not require changes in design or in material, and in fact it does not require processes that are fundamentally new or untried. It is merely the extension of processes that, on the whole, have long and honorable histories, and the avoidance of processes and practices that are now known to reduce fatigue strength. The significance of these processes has only recently become clear through the introduction of new concepts of fatigue phenomena by which new avenues of reasoning are opened to us. These new concepts are: *Fatigue failures result only from tension stresses, never from compressive stresses and any surface, no matter how smoothly finished, is a stress-raiser.*

■ Fatigue Vulnerability

The surfaces of repeatedly stressed specimens, no matter how perfectly they are finished, are much more vulnerable to fatigue than the deeper layers. It has long been appreciated that the vulnerability to fatigue increases as the surface roughness is increased, particularly if the roughness consists of sharp notches, and more particularly if the notches are oriented at right angles to the principal stress.

The practice of carefully finishing fatigue test specimens and engine parts is, of course, a recognition of this vulnerability in so far as visible marks or scratches are concerned, even down to being sure that the final polishing marks are parallel to the direction of the applied stress. These precautions are known to be effective in increasing the fatigue strength of the specimens, and specimens finished in this manner have, therefore, come to be known as "par" bars. This name implies that fatigue specimens and machine parts approaching perfection in finish give the highest possible fatigue endurance for any particular material, and that they accurately measure the ultimate fatigue properties of that material.

[This paper was presented at the SAE War Materiel Meeting, Detroit, Mich., June 9, 1943, and the National Aeronautic Meeting of the SAE, New York City, April 8, 1943.]

IT is doubtful whether we are getting more net work from metals today in dynamically loaded parts than was obtainable 25 years ago, and no super-strength-alloy discoveries seem imminent; however, much can be done to increase the fatigue strength of many machine parts made from ordinary structural materials by merely extending processes already known to be satisfactory, and avoiding practices that reduce fatigue strength.

We have today new concepts of fatigue failure: Fatigue failures result only from tension stresses, never from compressive stresses. Any surface, no matter how smoothly finished, is a stress-raiser.

Structural materials are not rigid. Many fatigue failures can be traced to elastic deflection for which no allowance was made in design.

From experience with practical machine parts, we can only conclude that stress calculations by textbook methods are wholly inadequate unless we generously temper our calculations with experience. The accuracy of stress data from photoelasticity, brittle lacquers, extensometers, and similar methods is usually greater than by mathematical analysis, but these are far from reliable.

As a working hypothesis, it seems reasonable to assume, except possibly for very ductile metals, that:

The slope of the fatigue curve, as measured on a log-log plot, is a measure of effective stress; and fatigue curves for varying stress concentrations converge toward a point near the tensile strength of the material at some considerable number of stress cycles.

Fully 90% of all fatigue failures occurring in service or during laboratory and road tests are traceable to design and production defects, and only the remaining 10% are primarily the responsibility of the metallurgist as defects in material, material specifications or heat-treatment.

Study of fatigue of materials is the joint duty of the metallurgical, engineering, and production departments. There is no definite line between mechanical and metallurgical factors that contributes to fatigue. This overlapping of responsibility is not sufficiently understood. Until more time is devoted to searching for mechanical causes rather than metallurgical ones, we cannot make full use of our materials.

THE AUTHOR: J. O. ALMEN has long been active in General Motors research development work. He is head of the Mechanical Engineering Department No. 1 of the GM Research Laboratories.

It can be shown, however, that the so-called "par" bars are not the best specimens, but that influences akin to

FATIGUE RESISTANCE

by J. O. ALMEN
Research Laboratories Division, General Motors Corp.

notches, so far as fatigue vulnerability is concerned, are retained by the "par" specimens. It seems that the specimen surface is highly vulnerable simply because it is a surface; that there is an extra hazard in the surface layer not shared by the deeper layers. This extra surface hazard may be due to submicroscopic notch effects, or to the fact that the surface is a discontinuity, since the outer crystals are not supported on their outer faces. Whatever the reason for surface vulnerability, the evidence of its existence is strong.

Fatigue Life Increased

The fatigue strength of the most carefully prepared specimen will be increased if a thin layer of the specimen is pre-stressed in compression[1] by a peening operation such as peen hammering, swaging, shot blasting or tumbling, or by pressure operations by balls or rollers. This increase in fatigue strength resulting from the surface layer being stressed in compression is clearly shown by the S-N curves, Fig. 1, which compare normally finished railway axles with axles that had been subjected to a rolling operation[2]. This and many other tests show that the compressive stressed surface is effective in increasing the fatigue strength whether applied to highly finished specimens or to specimens having rough surfaces.

We are all familiar with the improvement in fatigue that may be obtained by a few cycles of overload sufficient to produce a "set" in such parts as springs. Local tension stresses from the overloads exceed the elastic limit of the material and, therefore, the tension stress at the working load is decreased. This treatment, which has long been practiced on many production items, is similar in effect to rolling or peening since, in the unloaded state, the member is stressed in compression in the areas where yield occurred during the overloading.

The bar chart, Fig. 2, records the increased fatigue durability resulting from shot peening of a few typical machine parts. It will be seen that the fatigue durability is increased whether the parts are hard, such as carburized gears, or soft, such as steering-gear parts, and whether the stress is completely reversed, as in crankshafts, or the stress range is small, as in preloaded springs.

Note that the fatigue durability of peened axle shafts was not increased as much as most of the other specimens. The work on these shafts was conducted a number of years ago before the technique of peening machine parts had been developed, and the relatively small increase was probably due to insufficient peening. Similar fatigue results have been obtained from a large variety of machine parts and from aluminum specimens, and there are reasons to expect that the treatment is equally effective for all metals.

The bar chart, Fig. 2, shows the fatigue durability increase as per cent gain above the durability of the same machine part before peening. Actually, durability com-

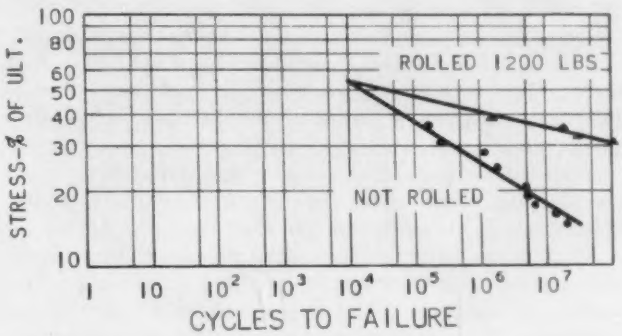


Fig. 1 – S-N diagram for railway axles showing change of slope due to rolling

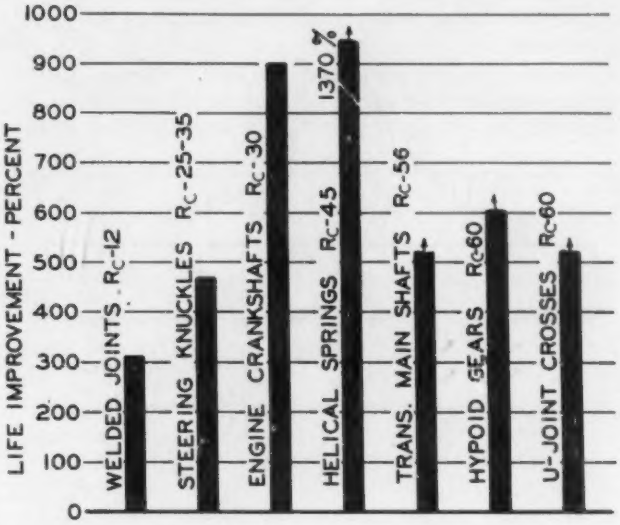


Fig. 2 – Bar chart showing the benefits of surface-peening based on fatigue life

parisons cannot be made on a percentage basis alone, as is apparent when we examine the improvement in fatigue due to rolling, as shown in Fig. 1. If, in this chart, the durability comparison is made at a load equal to 55% of the ultimate strength, the percentage improvement is zero; if the durability comparison is made at a load corresponding to 20% of the ultimate strength, the percentage improvement is infinite, and, at intermediate loads, the percentage gain will, of course, be somewhere between these limits.

It is essential that this be kept in mind when interpreting the fatigue data. To illustrate, suppose that the average fatigue durability of a gear tested at high load is increased from 30,000 cycles to 70,000 cycles by suitable surface-peening, a gain of 130%. Now if this comparison is made

[1] See *Stahl und Eisen*, Vol. 49, April 25, 1929, pp. 575-577: "Das Drücken der Oberfläche von Bauteilen aus Stahl," by O. Föppl.
[2] See ASME Transactions, Vol. 58, September, 1936, pp. A-91-A-98: "Increasing the Fatigue Strength of Press-Fitted Axle Assemblies by Surface Rolling," by O. J. Horger and J. L. Maulbetsch.

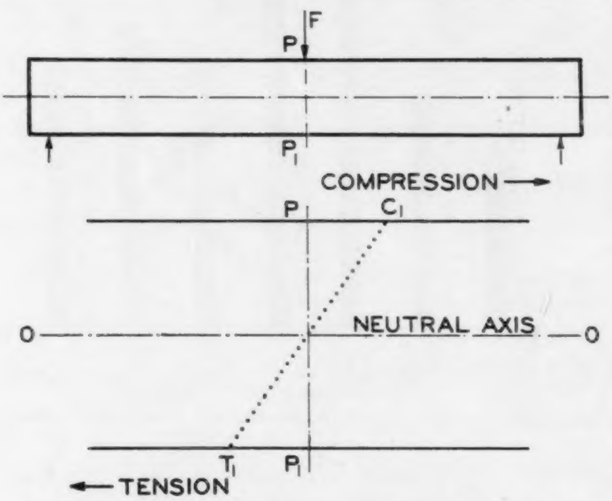
at a lower load such as to cause initial failure at 300,000 cycles, the treated gear might run 6,000,000 cycles before failure, a gain of possibly 2000%. This is because of the difference in slope of the fatigue curve representing the normal gears and the fatigue curve representing the peened gears.

■ Explanation of Peening Effectiveness

The most plausible explanation of the effectiveness of surface compression stress[3] is that when a load is applied to such specimens the tension stress in the surface layer becomes less by the amount of the compression pre-stress, and since fatigue failure starts from tension stress the fatigue durability of the weak surface layer is increased. However, the tension stress in the material below the pre-stressed layer is not reduced but may be actually increased, notwithstanding which, the fatigue strength of the specimen is increased. It follows, therefore, that the lower layer is inherently stronger than the surface layer.

Föppl[4] shows that the fracture in rolled specimens does not originate at the surface but in the material below the pre-stressed layer, as would be expected if the surface is sufficiently pre-stressed in compression. Similar subsurface fatigue failures, usually called fissures and attributed to faulty material, have long been known to occur in railroad rail, in which the surface is stressed in compression as a result of the cold work of heavily loaded locomotive and car wheels.

The situation can perhaps be clarified by the use of the conventional textbook stress diagram of a loaded beam, as illustrated in Fig. 3, in which a beam supported at the



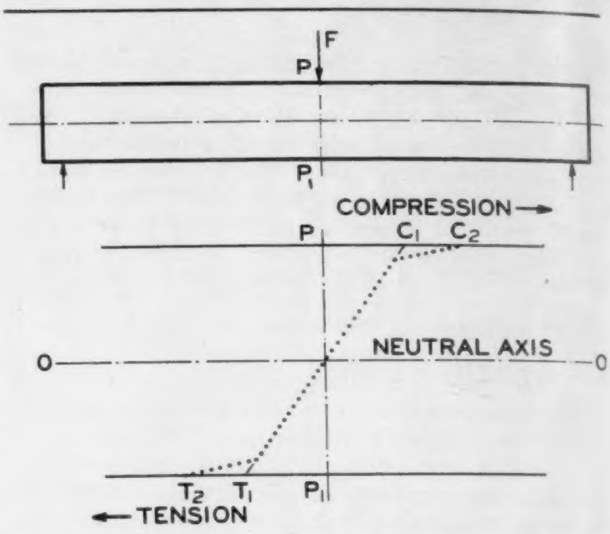
■ Fig. 3 – Conventional stress diagram of a loaded beam

ends is loaded in the central plane, P-P_1. The stress at any point in the beam is measured by the horizontal distance from the plane P, in which the load is applied, to the diagonal line T_1-C_1. The distance P-C_1 represents the compressive stress at the upper surface, the stress at the neutral axis O-O is zero and the tension at the lower surface is represented by the distance T_1-P_1.

[3] See *Stahl und Eisen*, Vol. 53, Dec. 21, 1933, pp. 1330-1332: "Die Wirkung von Eigenspannungen auf die Biegeschwingungsfestigkeit," by Hans Bühler and Herbert Buchholtz.

[4] See *Iron Age*, Vol. 126, Sept. 18, 1930, pp. 775-777, 829: "Cold Rolling Raises Fatigue or Endurance Limit," by G. S. von Heydekampf.

While this is a satisfactory enough stress diagram for static loads, it does not agree with the behavior of fatigue specimens. However, if we modify the diagram Fig. 3 as is shown in Fig. 4, in which T_1-T_2 represent an added

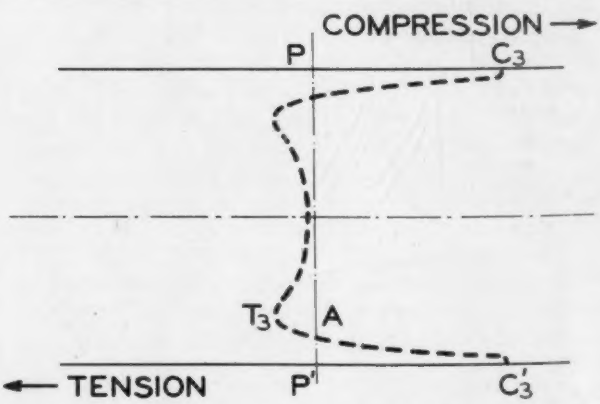


■ Fig. 4 – Modified stress diagram of a loaded beam showing surface vulnerability

increment of tension stress in the surface, we have a reasonable representation of the surface fatigue vulnerability. For a sharply notched surface, the additional stress increment T_1-T_2 is relatively great. As the surface roughness is decreased, the increment T_1-T_2 decreases, but no matter how well polished the specimen may be, there still remains an additional surface stress as measured by fatigue tests.

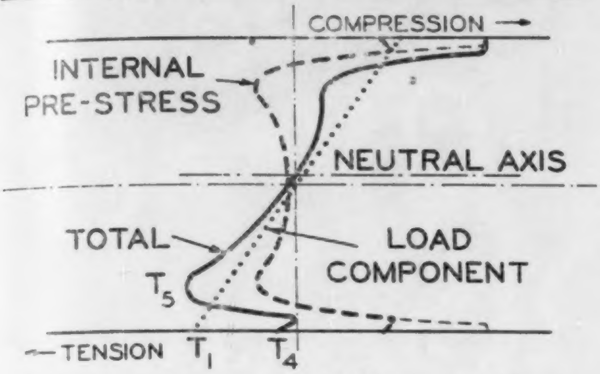
■ Stress Patterns

Fig. 5 represents the residual stress pattern in an unloaded beam that has been rolled or peened, as has been described, in which C_3-P and C'_3-P' represent the magnitude of compressive pre-stresses, and T_3-A represents the magnitude of the tension pre-stress to balance the compressed stresses in the surfaces. After this beam has been loaded from either side through one stress cycle, as in a reversed fatigue test, the compression pre-stress will be reduced if the applied load raises the total compression



■ Fig. 5 – Probable stress diagram of an unloaded beam with pre-stressed surfaces

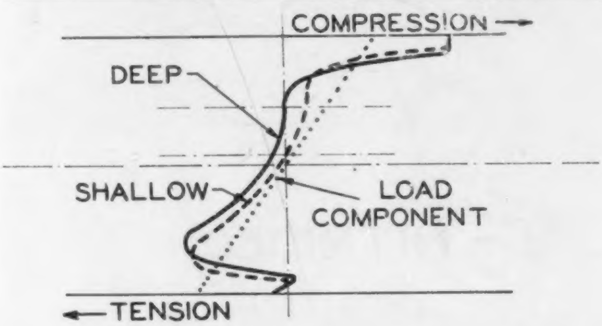
stress above the yield point. The stress diagram for such a pre-stressed beam supporting an external load is shown in Fig. 6, in which the effective tension stress, T_4, at the surface may be less than the stress T_5, below the surface,



■ Fig. 6 – Stress diagram for a loaded, pre-stressed beam

in which case failure would start below the surface, as noted by Föppl. Note also that the neutral axis is displaced from the geometric center of the beam and that the tension stress, T_5, below the surface is greater than in the beam that had not been pre-stressed, as is shown by the dotted lines.

The magnitude of the subsurface tension stress in a loaded beam having pre-stressed surfaces will vary with the amount of compression pre-stress and with the depth of the pre-stressed layer. Fig. 7 shows that the subsurface



■ Fig. 7 – Stress diagram for a loaded, pre-stressed beam showing the influence of the depth of the pre-stressed layer

tension stress may be greater for a deeply pre-stressed layer than for a layer of lesser depth.

It seems evident that the improvement in fatigue strength by compressive pre-stress is due to the reduction in tension stress when loaded in the vulnerable surface layer and that the increased compressive stress in a specimen stressed from zero to a maximum in either direction does no harm, probably because of adjustment of compression stress in the pre-stressed layer through yield.

Further evidence of the extra vulnerability of the surface layer is found in the behavior of specimens having increased strength in a thin surface layer, as in thinly carburized or cyanided specimens or in thinly nitrided specimens. Fatigue failures in such specimens also start below the surface and show greater fatigue strength than the same material in the unclad state. A nitrided specimen is probably superior to the other forms of hard cladding because, in addition to the higher physical properties of the surface layer, this layer is in a state of compression, and it is, therefore, less notch sensitive.

■ Residual Thermal Stresses

While on the subject of beneficial internal stresses, mention should be made of the surface compressive stress obtainable by heat-treatment. By a rapid quench, it is possible, through thermal contraction alone, to trap compressive stress in the surface and corresponding tension stress in the core, but this method, although showing some benefit in fatigue, is not as effective as the other methods that have been discussed. This subject will be discussed later in this paper.

Perhaps the most spectacular use of surface compression stress by heat-treatment through thermal contraction alone is tempered glass which, because of its great strength, is used in some parts of modern automobiles. This glass is prepared from normal glass by rapidly cooling the surfaces by means of air jets. The cooled surfaces contract causing the relatively plastic center to yield in compression. As the center of the glass cools and contracts it becomes stressed in tension, with consequent compressive stress in the surfaces.

■ First Use of Surface Compression

The idea of surface compression to improve the strength of steel is probably as old as steel itself. It has probably been discovered, forgotten and rediscovered many times. Certainly every village blacksmith knew and practiced the art in making wagon and buggy springs, axles, and other heavily loaded parts. After these parts were forged into shape they were severely hammered to improve their strength and, no doubt, the same procedure was followed by the ancient sword makers. Likewise, mill and ship shafts were cold-worked by the application of small rollers at high pressure after machining because of the greater strength that was known to result.

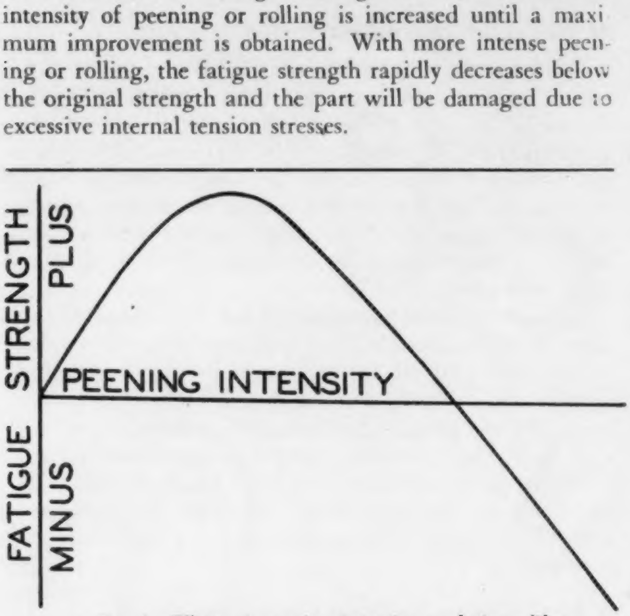
Cold-working of metals increases the hardness of most metals, including steel, at least in the range of low hardness, it usually results in internal stresses of varying degrees and patterns, it alters the physical properties and sometimes fractures the material. With the known sensitivity of materials to fatigue, it is obvious that we must learn how to control cold-work just as we have had to learn how to control heat-treatment in order that we may benefit by the good effects and overcome the evil effects. We would not think of specifying a heat-treatment without stating whether the temperature should be raised or lowered and in which order and to what extent; yet that is the way we now think of cold-work. Cold-working can be good or bad depending upon how it is done and for what purpose.

■ Pre-Stressing May Be Overdone

Papers have been published showing that cold-working of the surface so as to produce a layer stressed in compression increases the fatigue strength of the parts to which it is applied, but we are not told the amount of the pre-stress or the depth of the pre-stressed layer. Both of these values are presumably important in obtaining optimum results for any particular specimen, but it is probable that the values should not be the same for all sizes of specimens, for all materials, or for hard and for soft specimens.

Several instances are known in which the strength of machine parts and specimens has been decreased by too

intense surface peening. Fig. 8 is presented as showing the probable effect of peening or rolling, particularly on thin sections. The fatigue strength is increased as the intensity of peening or rolling is increased until a maximum improvement is obtained. With more intense peening or rolling, the fatigue strength rapidly decreases below the original strength and the part will be damaged due to excessive internal tension stresses.



■ Fig. 8 – Effect of peening intensity on fatigue life

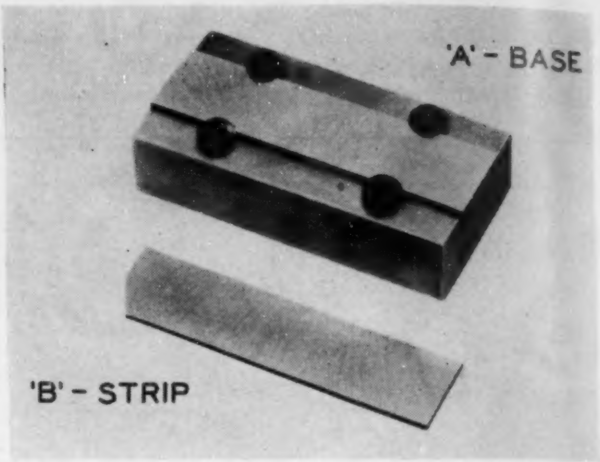
It would, therefore, seem important to control the compression stressed layer as to stress magnitude and depth with considerable accuracy by proper selection of the curvature of the rolling or peening instruments and by the pressure that is applied. The precise amount of surface compressive stress that is required for optimum fatigue strength is known for a few specimens only. It will vary with the shape and section thickness of the machine part, with the hardness, and with the kind of metal being treated. For the present, we must frequently rely upon the not too accurate sense of proportion that is developed by experience to indicate the treatment that should be applied to any given machine part.

When the layer is stressed in compression (by applying sufficient pressure on the work by rollers or by peening) to a degree exceeding the yield strength of the metal in compression, the amount of residual stress is presumably at least equal to this yield strength.

The depth of the stressed layer is probably roughly proportional to the instantaneous area over which the pressure is applied, and to the pressure intensity. The depth of the compression stressed layer in a railroad rail[5] should be greater than the depth of the compression stressed layer in the same material if small rollers at the same pressure intensity were used instead of large car wheels. Under these circumstances, the initial point of fracture should appear at corresponding depths. Such evidence as is available indicates this to be true.

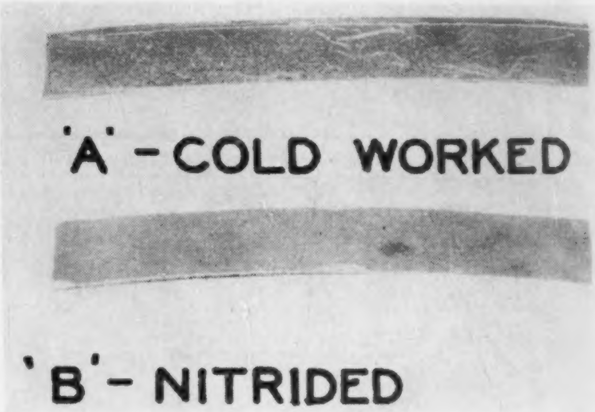
■ Instrument Measures Peening

A simple and practical method for indicating the compression in the stressed layer consists of a thin flat strip, Fig. 9B, that is attached to a heavy base as shown in Fig. 9A. This strip is rolled or peened with the same intensity that is given to the machine part, and when it

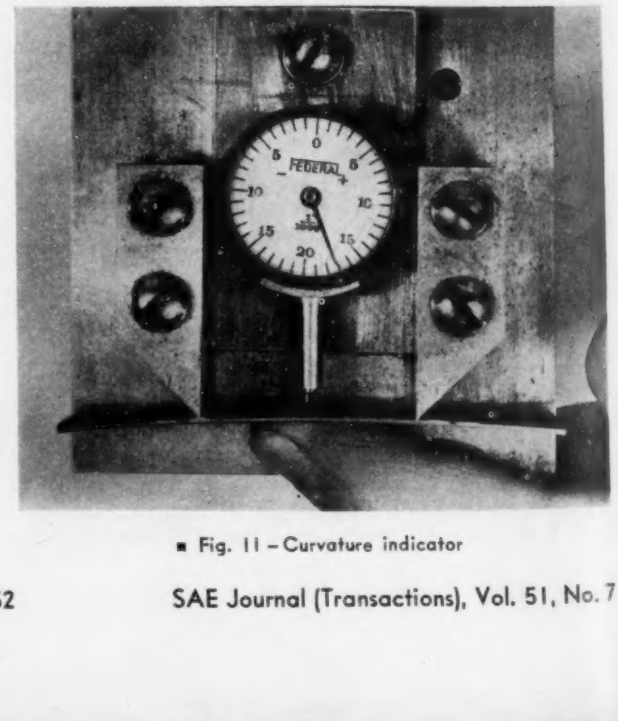


■ Fig. 9 – Apparatus for indicating the compression in the stressed layer

is removed from the base it will be found to be curved, as shown in Fig. 10A, with the convex surface on the cold-worked side. The curvature of the strip may be measured by an indicator, as shown in Fig. 11, which can then be interpreted in terms of the depth of the stressed layer.



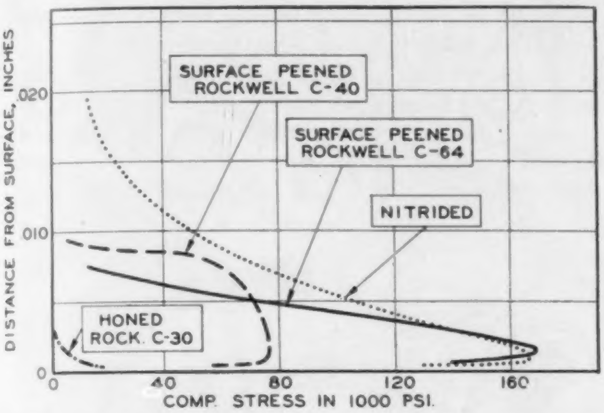
■ Fig. 10 – Measuring strips



■ Fig. 11 – Curvature indicator

[5]See *Journal of the Iron and Steel Institute*, Vol. 66, No. II, 1927, pp. 265-282: "The Work-Hardening of Steel by Abrasion," by E. G. Herbert.

The chart shown in Fig. 12 records the stress magnitude and the depth of the stressed layer at constant cold-work intensity of two such test strips. The cold-worked surfaces of these strips, the Rockwell C hardness being respectively 64 and 40, were honed away in small increments and the curvature was measured with the removal of each thin layer. The changing curvature as metal was removed provided data from which the compressive stress in each laver could be calculated, with the results shown in the chart. As would be expected, because of the higher yield point, the harder specimen was found to be more highly stressed than the softer specimen.



■ Fig. 12 – Magnitude and depth of stress imposed by various surface treatments

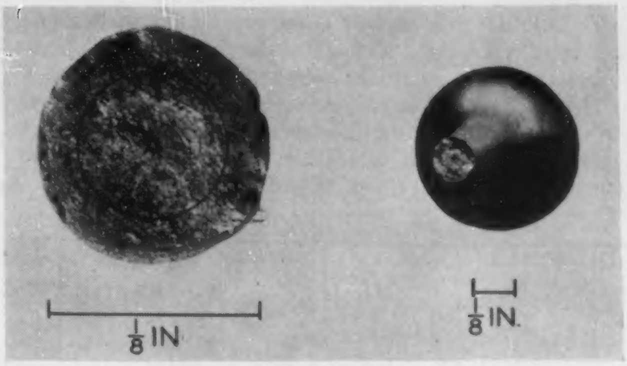
Also shown in this chart is the surface compressive stress in a nitrided specimen as a result of the nitriding. The procedure for this experiment was the same as for measuring the stress due to peening except that the face of the specimen that was in contact with the heavy base was plated to limit the nitriding to the outer face of the strip. On removal from the base after nitriding, the strip was curved convex on the nitrided side, as is shown in Fig. 10B. It seems, therefore, that the well-known resistance of nitrided specimens to fatigue is primarily due to the compressively stressed surface layer.

■ Overdose of Nitriding

Although the usual experience with nitriding is that it greatly improves fatigue strength, it is possible to overdo nitriding just as it is possible to overdo surface stressing by peening and rolling. The high compressive surface stress that results from nitriding must, of course, be balanced by internal tension stress of equal total value. When deep nitriding is applied to light sections, the unit internal tension stress may reach dangerous proportions.

Fig. 13 shows a part that was greatly reduced in strength as a result of nitriding, its fatigue durability being only 1 or 2% as great as the fatigue durability of the same part not nitrided. The diameter of the part at the point of failure was approximately ⅛ in. The depth of the nitrided layer was about 0.020 in., the area of which is equal to about 60% of the area of the section, as is shown by the circle in the enlarged view. From the nitriding, compressive-stress diagram shown in Fig. 12, it is evident that the internal tension stress must have been very great.

It is also known that internally nitrided cylinder barrels are more prone to fail by cracking than cylinder barrels that are not nitrided, the reason being that the stress due



■ Fig. 13 – The effect of deep nitriding – failure from severe tension stress

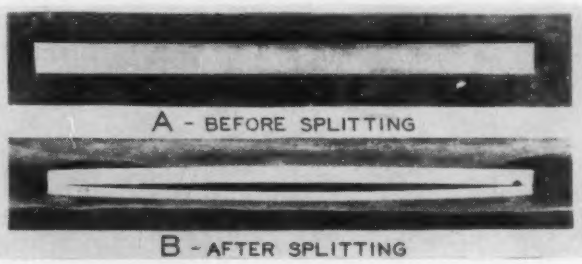
to nitriding is added to the stress from gas pressure. Care must, therefore, be used in nitriding thin sections to gage the depth of the nitrided layer in proportion to the thickness of the section being nitrided.

■ Residual Stress from Honing

While the peened specimens used for the experiment shown in Fig. 12 were being honed as has been described, it was found that the strips did not fully recover their original flat form. To determine if this residual curvature was due to a "set" in the material or was the result of honing, other flat strips that had not been peened were honed. These strips developed the same curvature as the residual curvature in the peened specimens, demonstrating that honing produces a compressively stressed layer. The approximate magnitude of this honing stress is also shown in the chart given in Fig. 12. This raises a question as to the state of surface stress in the carefully prepared fatigue specimens favored for laboratory fatigue tests, since additional tests have shown that lapping also introduces surface compressive stress.

■ Residual Stress from Carburizing

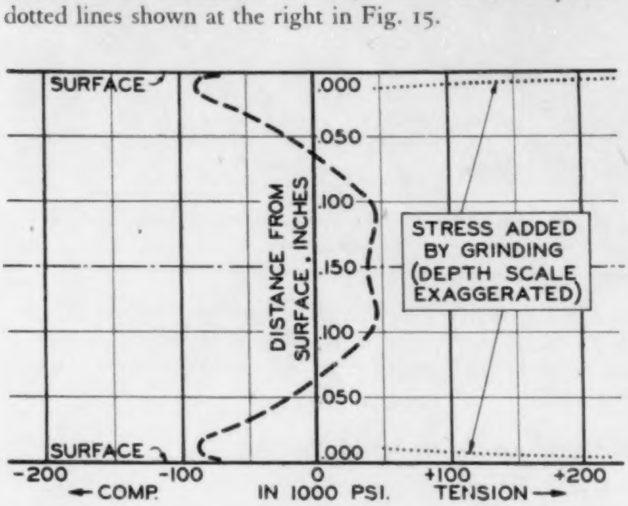
The carburized layer in a carburized part is stressed in compression, as is graphically shown in Fig. 14. Two



■ Fig. 14 – Residual compressive stress resulting from carburizing and hardening the upper and lower faces of a specimen

opposite faces of this ½-in. square specimen were carburized, while the other two faces were protected by copper plating. The specimen was quenched and tempered in the usual manner, after which it was split with a saw as shown in Fig. 14B. Note that the parts are curved convex on the outer faces, indicating compressive stress in these faces. Analysis of the internal stresses in another carburized member by a method similar to that described for peened and nitrided strips indicated the internal stress pattern shown

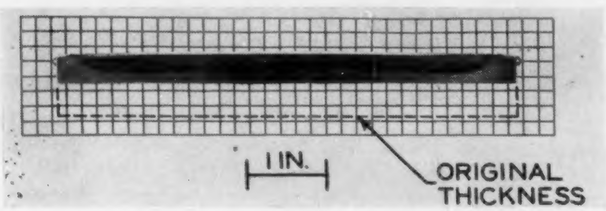
in Fig. 15. Of interest here is the magnitude of the compressive stress in the carburized layer and the reduced compressive stress, possibly even tension stress, in a thin surface layer. When carburized parts, such as bearing races, wristpins, and gear teeth, are ground we may expect the surface to be stressed in tension, as is indicated by the dotted lines shown at the right in Fig. 15.



■ Fig. 15 – Magnitude and depth of residual stress due to carburizing and hardening

The compressive stress in the carburized layer may be a hazard for members stressed in tension, as was shown for the nitrided part, Fig. 13, because the tension stress in the core is equal to the working load plus the tension load due to the compressive pre-load of the case. For members stressed in bending and in torsion, however, the internal compressive stress in the carburized case of ordinary depth improves the fatigue strength of the part except for the thin surface layer which, especially after grinding, is severely stressed in tension. It is, however, a simple matter to convert this thin tension stressed layer into stress in compression by suitable peening or rolling operations, as was indicated in Fig. 12, with resultant large gains in bending and torsion fatigue strength.

The residual stress in crankshafts and other parts hardened by induction heating and probably also in flame-hardened parts resembles the residual stress in carburized and hardened parts, as shown in Fig. 16. The hardened



■ Fig. 16 – The effect of induction hardening – the convex curve of the upper surface indicates compressive stress

upper surface of this specimen was straight at the original thickness. Note that after removal of the material indicated by the dotted line, the upper surface is curved convex, indicating compressive stress. More complete analysis indicated that a thin surface (upper) layer was possibly stressed in tension. In these treatments as in carburizing, the hardened layer is stressed in compression because in undergoing the phase change to the hard state, the density of the steel is reduced and therefore the hardened layer seeks to occupy more space. A thin surface layer however may be stressed in tension.

With internal stresses of the magnitude shown in Fig. 15, we can readily understand why carburized parts are prone to warp during heat-treatment, especially if the design is not symmetrical with respect to the internal stresses.

■ Low-Temperature Quenching Stresses

Residual stresses due to quenching from relatively low temperatures may reach considerable magnitudes and may be harmful or helpful to fatigue durability depending upon whether the trapped stresses augment or diminish the tension stresses from the applied loads. An interesting case of this kind occurred in a water-cooled aluminum cylinder head, as shown in Fig. 17, that failed by fatigue

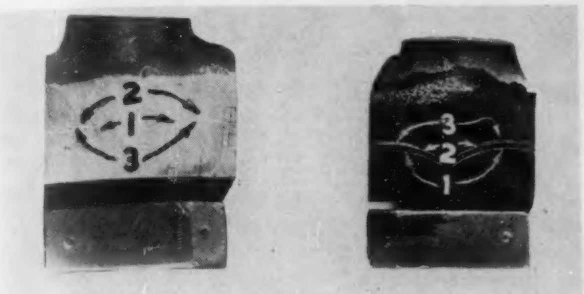
■ Fig. 17 – Method of indicating quenching stresses in a water-cooled aluminum cylinder head – fatigue failure occurred on the water side of the combustion-chamber wall

on the water side of the combustion-chamber wall. Measurements of residual stress disclosed that the water side of the combustion-chamber wall was stressed in tension and the combustion-chamber side of the wall was stressed in compression. This internal stress pattern was one of the same kind as the stress from the gas pressure against the combustion-chamber wall, and the resultant stress was, therefore, the sum total of the residual stress and the gas pressure stress.

The residual stresses in this case were caused by quenching the cylinder heads from 980 F by immersion in cold water. The outer surfaces of the casting were cooled while the inner water-jacket surfaces, especially at the thick section, were still hot. Thermal contraction of the outer surfaces imposed compressive stresses of such magnitude as to cause yield in the still hot and therefore weaker water-jacket surface. As cooling progressed, the metal that had been stressed beyond the yield point contracted thermally, leaving tension stress on the water-jacket side and the corresponding compression stress on the combustion-chamber side. The retention of the residual stress in the thick combustion-chamber wall was aided by the thinner outer wall of the water jacket, which was stressed in compression.

A visual indication of the residual stress pattern is shown

by the scribed arcs at the left in Fig. 18. These arcs were drawn from centers in a steel bar bolted to the opposite side of the casting section, as is shown in Fig. 17. The arcs indicated by the numeral 1 were drawn when the cylinder head was intact. The casting was then sectioned as shown in Fig. 17 except that the outer side of the water jacket had not been cut and the arcs indicated by the numeral 2, of the same radius as before, were drawn. Finally the outer side of the water jacket was removed and the arcs indicated by the numeral 3, still of the same radius, were drawn. Note the direction of movement of



▪ Fig. 18 – Variation in internal stresses from different quenching methods

the metal with each operation and the order of the stress that is indicated. As a correction of the undesirable residual stress shown by this test, another cylinder head was given the same heat-treatment except that it was quenched internally by forcing cold water through the water jacket. The order of internal stresses was measured in the same manner as has been described, with the results shown at the right in Fig. 18. Note that when internally quenched, the stress pattern in the combustion-chamber wall is reversed, leaving the inner side in compression and the outer side in tension. Since these trapped stresses are of opposite sign to the operating stress, the resultant stress is the difference between the residual and operating stresses instead of their sum as when the casting was quenched externally.

■ Fatigue Life Increased

Fatigue tests were conducted on cylinder heads quenched by both methods, the results of which were 2,000,000 to 3,000,000 stress cycles to failure for the externally quenched heads, and 5,000,000 to 6,000,000 stress cycles for the internally quenched heads.

Additional fatigue tests were made on internally quenched heads in which the aging treatment at 350 F was omitted in order to avoid reduction of the favorable stress pattern. These heads endured more than 14,000,000 stress cycles at the same test load without failure.

Similar residual stresses are known to occur in many other heat-treated and quenched aluminum parts. It is also known that many aluminum parts show better fatigue resistance when they are drawn to a higher temperature than that which gives the greatest tensile strength, presumably because unfavorable residual stresses resulting from quenching are thereby reduced. It is probable that similar stresses can be trapped in steel by quenching from tempering temperatures[6]. Such residual stresses may be favorable or unfavorable depending on the shape of the part, the temperature gradient, and the direction of heat flow.

■ Corrosion Promotes Fatigue

Fatigue failures in many machine parts are traceable to corrosion of several kinds or to other forms of surface damage that occur in service. In normal machine parts, even slight corrosion or bruising is very potent in encouraging fatigue fractures because each pit interrupts the continuity of the surface and increases the local stress. The damaging effect of corrosion or bruising is prevented on the surfaces that are adequately protected by compressive pre-stress because the local tension stress cannot reach dangerous values until the pits or bruises have progressed sufficiently to penetrate the compressively stressed layer. This was forcefully demonstrated in fatigue tests of a machine part that failed alternately in a badly formed fillet or in the region of a clamp remote from the fillet where fretting corrosion occurred. The durability of the part could not be increased by improving the fillet because this would merely transfer all failures to the fretted area at about the same durability. After peening, however, the fatigue durability was found to have increased several hundred per cent and large additional gains were then possible by improving the form of the fillet without failure in the corroded area. The peening did not prevent corrosion but it did prevent the ill effect of corrosion in promoting fatigue.

Similar protection against the effects of corrosion and of surface bruises is afforded by nitriding[7], carburizing, and other treatments that produce compressively stressed surfaces. The working face of a gear tooth may be severely pitted, creating a fatigue hazard, but the bending fatigue strength may not be impaired because the carburized layer is compressively stressed and the surface is compressively stressed by the cold-work of mating teeth.

■ Surface Finishes

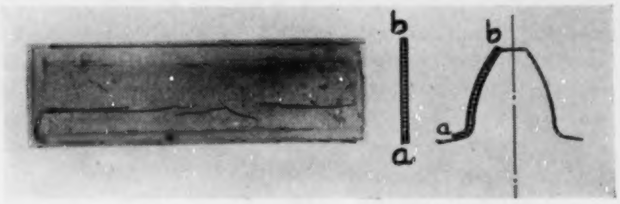
Efforts to improve products by improving surface finish may sometimes have the opposite effect. Highly finished surfaces and fillets may lead to a false sense of security if, as the result of machining or straightening operations, the parts have high internal stresses of the wrong kind.

When machine polishing is done by the use of abrasive paper, cloth wheels, or abrasive-covered felt wheels, sufficient heat is often generated to induce serious surface-tension stresses and thus promote instead of prevent fatigue failures.

In ground surfaces such as shafts, wristpins, and gear teeth, the grinding operations may introduce high surface tension stresses that, from the standpoint of fatigue strength, often do more harm than good. The surface-tension stresses from grinding are often so great as to produce visible or magnaflux surface cracks, but whether detectable or not, surface tension is frequently very serious.

Fig. 19 is a magnaflux transfer print on transparent cellulose tape showing surface fractures in a ground gear tooth. This tooth failed by spalling originating in these surface fractures. Since fatigue cracks start on the side of the gear tooth that is loaded in tension, the effective stress is the grinding pre-stress plus the working stress. Fre-

[6] See *Engineering*, Vol. 154, Aug. 14, 1942, pp. 134-135: "Quenching of Steel after Tempering and the Impact Test," by L. E. Benson.
[7] See *Metals and Alloys*, Vol. 5, June, 1934, pp. 129-130: "Effect of Notches on Nitrided Steel," by J. B. Johnson and T. T. Oberg.

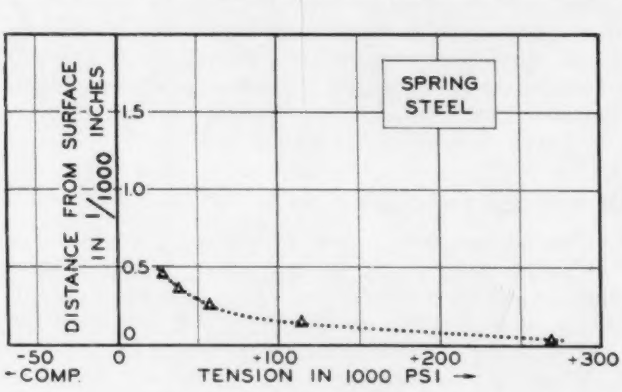


■ Fig. 19 – Surface fractures in a gear tooth – the result of grinding

quently, we find that a hardened part will show a file-soft skin after grinding, which not only promotes fatigue but is also susceptible to seizure and galling.

Internal stresses of the wrong kind are perhaps the most insidious of all fatigue hazards because we can seldom know their magnitude or the pattern in which they are distributed within the material or whether they are alike for all commercially identical machine parts. Internal stresses may be the result of operating conditions such as occur in brake drums, clutch plates, or other friction surfaces where the instantaneous temperature in a thin layer is so great that, under thermal expansion, the surface layer is stressed beyond the yield point in compression. When the source of heat is removed, the heated surface layer is quenched by the adjacent cool metal and, under thermal contraction, it is so severely stressed in tension that fractures often occur. This is, of course, the same thing that happens in machine polishing and in grinding unless great care is used.

The magnitude of surface tension stress in a specimen that was ground in accordance with normal commercial practice is shown in Fig. 20. A specimen of annealed



■ Fig. 20 – Magnitude and depth of the residual stress caused by grinding a spring-steel specimen

spring stock, 1/16 in. in thickness, 1 in. wide, and 7 in. long was ground to a depth of 0.002 in. After grinding, the previously straight specimen was found to be curved concave on the ground side, indicating tension stress. Very thin layers were then removed from the ground surface by hand honing until the specimen regained its initial straightness. Measurements of the change in curvature with each thin layer removed permitted calculation of the stress distribution as is shown in the chart. Surface stresses of this magnitude are not unusual in ground production parts, but we are seldom aware of their presence unless actual failure has occurred.

Obviously, a stress of 270,000 psi, a stress just below the fracture point of full-hard steel, could not be supported by the steel in the annealed state, from which it follows that the stress layer was hardened by the heat cycle of the grinding operation to not less than Rockwell C 55. The extreme thinness of the hardened layer presents an interesting problem in hardness measurements, as is shown in Table 1.

TABLE I

	Unground	Ground
Rockwell B	88	89
Rockwell C	5	5
Vickers Brinell	193	199

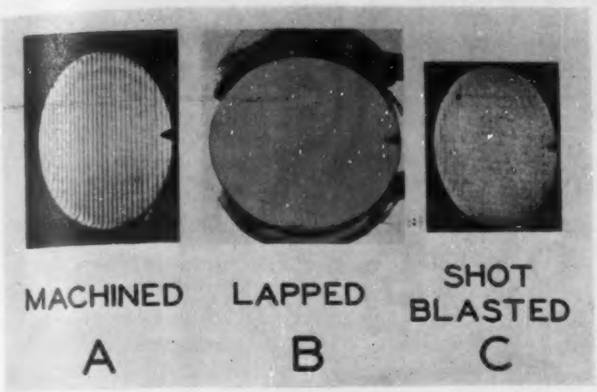
This table demonstrated the futility of our normal hardness-measuring technique for measuring the hardness of the most significant portion of our machine parts, the surface layer.

Internal stresses often result from the cooling of castings and forgings or from the vigorous heat transfer of heat-treating. Many parts, such as crankshafts, axle shafts, and camshafts require straightening during processing. Since the straightening operation is usually done at room temperature and since the part is rarely stress relieved after straightening, the result is severe internal stresses.

■ Machining Damage

In turning, milling, and other machine operations, it is necessary that metal be removed at a minimum cost, and therefore the cutting tools must often take deep cuts at high feed rates. Since metal cutting is more accurately described as a metal-tearing operation so far as stresses are concerned, we need not be surprised to find serious internal stresses to considerable depths after machining. When metal cutting has been unusually severe or after operations such as punching and shearing, we often find that the surfaces are actually fractured. Finish machining or grinding rarely goes deep enough to remove the internally stressed metal from previous rough machining and, of course, these finishing operations add stresses of their own. Whenever it is economically practicable, internal stresses that produce tension in any surface layer subjected to cyclic tension stress should be reduced or removed or, better still, converted to compressive stress by suitable treatment because all fatigue failures are due to tension stresses.

In connection with machining damage, an interesting and perhaps important observation has recently been made which indicates that the layer "injured" by machining is deeper than is generally believed. It also shows that the "injured" material does not recover by heating for long periods at high temperatures. Fig. 21A shows a bar of 4615 steel as it appeared after rough machining on a shaper. This piece was then carburized for 8 hr at 1700 F, cooled in the box, reheated to 1500 F, quenched in oil, and drawn at 300 F for 1 hr. The machined surface was then ground in a direction at right angles to the shaper marks to a depth of 0.0055 in. below the last visible tool mark, after which it was polished as shown in Fig. 21B. Finally, the polished surface was shot blasted, whereupon the machining marks (vertical lines) and the grinder marks (horizontal lines) reappeared as shown in Fig. 21C, showing that the material is not uniform in resisting the shot blasting, notwithstanding the long period at elevated temperature. There is no evidence at present that the effect brought out by this experiment is significant in fatigue.



▪ Fig. 21 – A steel part, rough machined as shown in *A*, after being heated at a high temperature for a long period, was ground and polished smooth, as shown in *B* – shot blasting the polished surface again brought out the machining marks, as shown in *C*

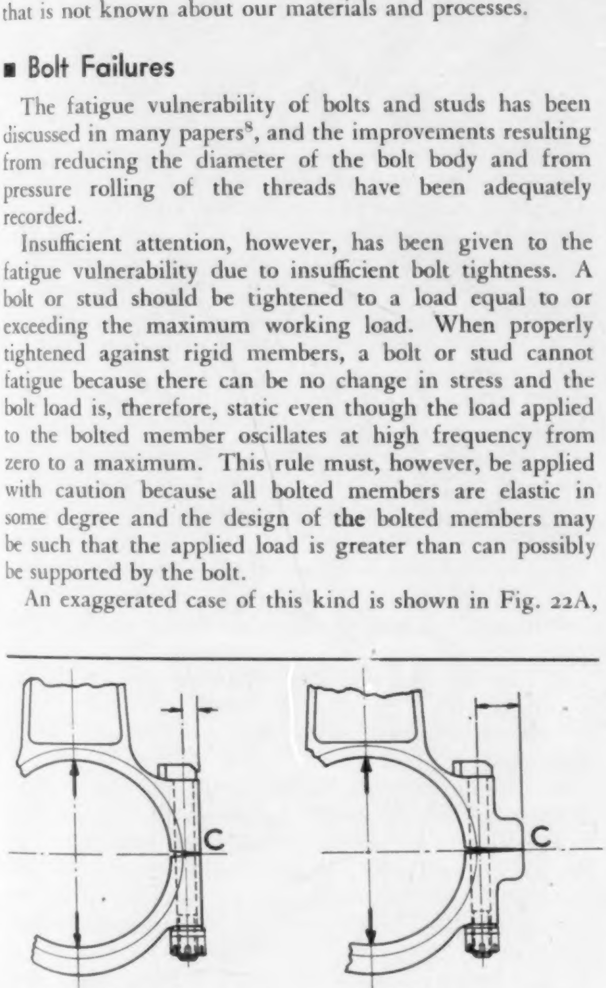
It is presented here merely to emphasize that there is much that is not known about our materials and processes.

▪ Bolt Failures

The fatigue vulnerability of bolts and studs has been discussed in many papers[8], and the improvements resulting from reducing the diameter of the bolt body and from pressure rolling of the threads have been adequately recorded.

Insufficient attention, however, has been given to the fatigue vulnerability due to insufficient bolt tightness. A bolt or stud should be tightened to a load equal to or exceeding the maximum working load. When properly tightened against rigid members, a bolt or stud cannot fatigue because there can be no change in stress and the bolt load is, therefore, static even though the load applied to the bolted member oscillates at high frequency from zero to a maximum. This rule must, however, be applied with caution because all bolted members are elastic in some degree and the design of the bolted members may be such that the applied load is greater than can possibly be supported by the bolt.

An exaggerated case of this kind is shown in Fig. 22A,



▪ Fig. 22 – Connecting-rod design – *A* causes excessive tension stress on the bolt, *B* is an improved design

in which the bolts are excessively stressed in tension and in bending because the distance from the bolt to the point *C* is small, and since the bolted parts tend to bend about the point *C* as a fulcrum, the tension and bending loads in the bolts are great. Fig. 22B illustrates an improved design in which the fulcrum point *C* is farther removed from the bolt and, therefore, the tension and bending loads are reduced. This is a case in which fatigue failure of one member is due to faulty design of another member, as is frequently encountered in practice.

If the bolt in Fig. 22B should fail by fatigue, the failure could still not be charged to insufficient bolt strength because, as stated, if the initial bolt tension is less than the applied load, the stress range under repeated loads is increased. Let us suppose that the bolt is tightened just enough to bring the surfaces into contact without appreciable tension. Under alternating stress, the stress range would be from zero to maximum and fatigue failure could only be avoided by greatly increasing the bolt strength. As the initial bolt tension is increased, the stress range is decreased until it approaches zero when the initial bolt tension is equal to or greater than the maximum working load. This illustrates a case in which a bolt fatigue failure is not the fault of either bolt strength or of design but is chargeable to bad assembly practice.

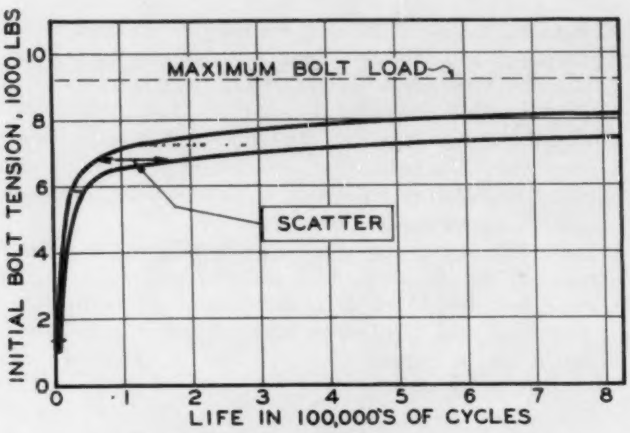
The vulnerability to fatigue as a function of bolt tightness is shown in Fig. 23. In these tests, all bolts were subjected to a cyclic tension load of 9210 lb, but were tightened to initial tensions of 1420, 5920, 7220, and 8420 lb. Fifteen bolts were tested in each of the three lower groups as shown in the graph in order to establish partially the scatter band for this kind of specimen. Only two bolts were tested in which the initial tension was 8420 lb, one of which failed after 4,650,000 stress cycles and the second was not failed after 10,000,000 stress cycles. These are not shown on the graph because they would compress the scale to undesirable proportions. The bolts used in these tests were ⅜ in. in diameter accurately dimensioned and finished. The threads were uniform, 24 threads per in. and ground to close limits.

▪ Stress Range in Bolts

The stress range to which these bolts were subjected is the difference between the initial load and the maximum operating load, and since it is known that the fatigue durability is increased as the stress range is decreased, we would expect results of the order that were obtained from these tests as shown in the chart. All failures occurred in the threads except in a few cases in which the threads were rolled in a manner to pre-stress the roots of the threads in compression. In these rolled threaded bolts, the fatigue durability of the threads was increased sufficiently to cause failure in the bolt shanks. When the surfaces of the bolt shanks were also compression pre-stressed by peening, the failures were again transferred to the threads but, of course, at prolonged durability. These tests, therefore, also show that the fatigue durability of cut and ground screw threads can be increased by rolling, and indicate that compression pre-stressing of the surface of pure tension members is effective in increasing their fatigue strength.

It is evident that the fatigue strength of bolts and studs stressed in tension is dependent upon the initial tension applied by the nut and upon the elasticity of the bolted members. Therefore, washers, lock washers, gaskets, and other units that add to the elasticity of the bolted assembly are definite fatigue hazards and should be avoided wherever possible. Short studs or bolts are more vulnerable to

[8] See "Prevention of the Failure of Metals under Repeated Stress," by Battelle Memorial Institute, John Wiley & Sons, Inc., 1941.

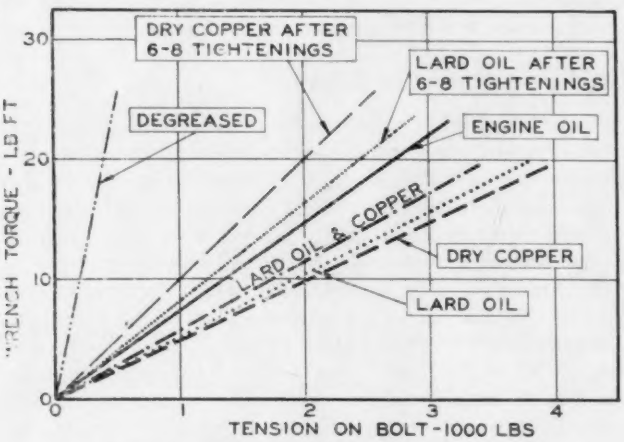


■ Fig. 23 – Effect of initial tightness on bolt life

fatigue than long ones. Their elastic extension is small, and therefore a slight loss of dimension by corrosion, plastic deformation, or wear will lose preload and they will fail by fatigue.

The practice of plating screw threads with soft metal to avoid corrosion is a definite fatigue hazard and should be avoided. The soft plate is so weak that it squeezes out from between the loaded thread surfaces and reduces the stud or bolt tension, thus promoting fatigue failure. A slackening of 0.001 in. on a stud holding a ½-in. flange may result in a loss of 50,000 psi preload. If protective plating must be used, it should be of a hard metal and of minimum thickness.

The initial tension applied by the nut is difficult to determine unless the elongation of the bolt or stud can be measured. Measurement of the torque applied to the wrench is very unreliable because of the variability of the friction. Fig. 24 records tension measurements plotted



■ Fig. 24 – Effect of lubricants on bolt tension

against wrench torque in lb-ft for 5/16-in. diameter capscrews having 24 threads per in. It will be seen that the bolt tension varied as much as 10 to 1 for constant wrench torque depending upon the lubricant that was used. The mechanical efficiency of this bolt varied from 1 to 10%, as may be calculated from the chart.

There is little need for metallurgical examination ot failed bolts or studs, or for considering design changes until it has been shown that the failure was not the fault of the man on the wrench.

Preloading of cyclically stressed members to reduce the stress range and thus to increase their fatigue durability is not restricted to bolts but may be applied to many machine parts. For example, the stress range in leaf-spring eyes can be reduced by pressing a bushing tightly into the spring eye.

■ Materials Are Elastic

A common cause of fatigue vulnerability is the belief apparently held by many designers and engineers that our structural materials are rigid. Many fatigue failures can be traced to elastic deflection for which no allowance was made in the design. Elastic deformation of mating parts may be such as to concentrate the load in a small region, as occurred under the conditions described for the bolt in Fig. 22A.

Under operating conditions, a crankshaft may be so elastically deformed in twisting and in bending that the bearings are only partially effective in supporting the load. The bearings are frequently found to be plastically deformed or worn "bell mouthed" to accommodate the elastic gyrations of the crankshaft.

Perhaps the most generally misunderstood of all machine elements are the several classifications of gears. As ordinarily designed, there is only one thing certain about gears and that is that they will not function as intended by the designer. When laying out a set of gears on the drafting board, the mating gear teeth are represented by parallel straight lines over which the load is assumed to be uniformly distributed, but no matter how carefully the gears are cut and heat-treated the mating teeth will never again be parallel except by accident and then only through a small load range.

The nature of the contact between two mating gear teeth is influenced by:

1. The elastic characteristics of the housing in which they are contained.

2. The elastic characteristics of the bearings by which they are supported.

3. The elastic characteristics of the shafts upon which they are mounted.

4. The elastic characteristics of the gears themselves.

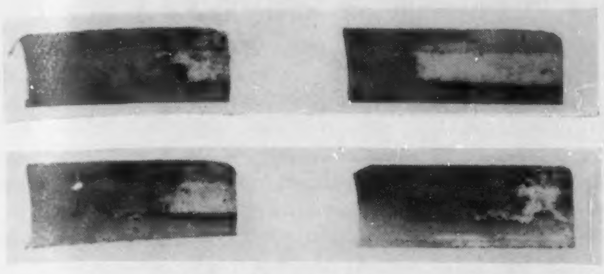
5. The accumulated dimensional errors in all the supporting parts as well as the errors in the cutting of the gears.

6. The necessary and accidental clearances in the supporting parts.

7. Misalignment of supporting parts through thermal expansion.

8. The amount and nature of the warpage in heat-treating – to give the metallurgist some of the responsibility.

The result of all this is that it is virtually impossible for the parallelism between mating teeth, as visioned by the designer, to exist in practice. If it should chance that two mating gear teeth are parallel at some load, they cannot be parallel at any other load because the elastic deflection of some of the supporting parts is not linear with respect to the load. As ordinarily designed, the load on gear teeth is never uniformly distributed over the length of the teeth but is always concentrated toward one end of the teeth. This localization of the load is shown in Fig. 25, which is a record of the contact impressions of gear teeth under load in a commercial gear box. Load localization cannot often be seen by examination of a gear that has been in service because, usually, each tooth of each gear makes



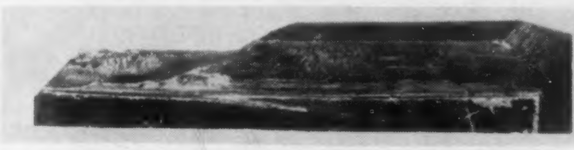
■ Fig. 25 – Contact impressions of gear teeth under load in a commercial gear box

contact with all of the teeth in the mating gear and, therefore, the summation of all contacts under all load conditions will be seen by the examiner.

■ Localized Gear-Tooth Load

The illustration shown in Fig. 26A is of a gear that failed in service. This gear was "rescued" while on its way to the metallurgical department to find out what was wrong with the material to cause the fatigue failure. Note that the failed tooth is broken at one end which, incidentally, is typical of almost all failed gear teeth. An adjoining unbroken tooth shown in Fig. 26B tells us that failure

■ Fig. 26 – Localized tooth load of a gear that failed in service



A – Gear tooth that failed in service

B – Nature of contact, as shown by adjoining unbroken tooth – load was concentrated on left end

occurred because only a small part of the tooth was actually supporting the load in spite of the generous tooth length that was provided by the designer. This gear would have been just as durable had it been designed to one-fifth the tooth width that was actually provided. Clearly, this is a mechanical and not a metallurgical problem. The real trouble was inadequate support of the gears and other mechanical errors as enumerated.

It may fairly be argued that this is an unusually severe case and that it is not typical of gear fatigue. Actually the most unusual thing about it was that it could be diagnosed before it was cut into sections and the evidence etched away, as so often happens in metallurgical examinations.

In the case of fatigue failure of mating helical gear teeth of equal strength, fatigue will always occur in the tooth that is loaded on its acute-angled end because the section is weaker at this end. Mating helical gears should be offset so that contact cannot occur on the acute-angled end by any mode of deflection. This is possible only where the torque is constant in direction.[9]

All gear teeth should be designed to afford a degree of tolerance for deflections, machining errors, and warpage as has long been standard practice in spiral bevel[10], hypoid, and in some spur and helical gears. In "rigidly" mounted gears, this is accomplished by curving the teeth barrel shape or the equivalent in such a manner as to concentrate the load near the centerline of the gear width and thus avoid load concentration at the weaker extreme ends of the teeth.

Load concentrations at the ends of gear teeth can sometimes be avoided by increasing the elastic deflection, as is done in simple spur reduction gears by providing the gear with a thin diaphragm web located central of the gear width, or by more complex construction in other forms of gears, but it cannot be done by accurate dimensions alone.

■ Gear-Tooth Pitting

The pitting of gear teeth is a form of fatigue that is induced by tensile stress from compression loads on the contacting tooth surfaces. The magnitude of the compression stress varies with relative curvature of the contacting teeth in accordance with the Hertz formula, it varies with the degree of load concentration at the ends of the teeth and with the applied load. The load that may be carried varies with the hardness and, therefore, with the strength of the material, with the temperature, and with the manner in which the lubricant is applied.

The design factors that are effective in reducing the load concentration at the ends of the tooth barrel shaping or equivalent, also reduce the contact compressive stress. The relative curvature and, therefore, the contact compressive stress can be varied by the choice of pressure angle. In general, there is little to be gained by designing wide face gears except the doubtful satisfaction of dealing with the smaller stress numbers.

In high-speed gears, pitting may occur when gears are transmitting no load. This is sometimes seen in the reverse idler gear of the automobile transmission. Although this form of transmission trouble is rare and occurs only when other conditions, such as hardness, are unfavorable, it serves to emphasize the part played by the lubricant in promoting fatigue. A reverse idler running submerged in oil will trap the oil between the gear teeth and if the clearances are small will induce extremely high surface pressures. We are all familiar with the high temperatures that are generated in gear boxes when too generously supplied with oil, but we do not always interpret this as a fatigue hazard. High-speed gears should be lubricated by jets of low-viscosity oil directed at the teeth as they are coming out of mesh, not on the incoming side. This form of lubrication will wash away the heat of friction while it is still on the surfaces of the teeth and will prevent excess oil from reaching the contacting teeth, provided, of course, the sump is dry.

■ Generalized Fatigue "Laws"

The conventional approach to studies of the fatigue of metals is through laboratory tests on several arbitrary forms of fatigue specimens. During the many years that such tests have been made, a vast amount of fatigue data have been accumulated. These data have enabled us to

[9] See *Automotive Industries*, Vol. 77, Sept. 25, 1937, pp. 426-432, Oct. 9, 1937, pp. 488-493: "Factors Influencing the Durability of Automobile Transmission Gears," by J. O. Almen and J. C. Straub.

[10] See ASTM Proceedings, Vol. 35, Part II, 1935, pp. 99-135: "Rear Axle Gears; Factors which Influence their Life," by J. O. Almen and A. L. Boegehold.

formulate somewhat generalized "laws" on the behavior of various specimens subjected to repetitive stresses of several kinds.

We have found that steel, under most laboratory conditions of repetitive stress has a fairly well-defined limit of stress, known as the fatigue endurance limit, below which it will endure for an infinite number of stress cycles; that the fatigue endurance limit of steel is roughly proportional to the ultimate strength of the material but that the proportionality varies with the range of the applied stress. We also know that, under certain other test conditions, steel does not have a fatigue endurance limit, that nonferrous metals generally do not have a fatigue limit, that rough surfaces, notches, section changes, and other discontinuities are detrimental to fatigue strength. These and many other "laws" have been established through laboratory tests under controlled conditions.

The preferred laboratory fatigue test specimen is very carefully prepared to avoid all surface imperfections, abrupt section changes, internal stresses, and other stress-raisers. This is considered necessary because the investigator is usually interested in the inherent properties of the material undergoing test, and he naturally seeks to eliminate all factors that would tend to obscure these inherent properties. There can be no objection to this procedure as it refers to the test specimens, but the data thus obtained have little bearing on the fatigue characteristics of machine parts made from the same material and given the same heat-treatment, because in machine parts surface irregularities, abrupt changes in section, and internal stresses are almost always present.

Economic Requirements

In the design of machines and equipment for heavy duty, where weight is not important and where the number of units produced is small, the present practice of designing to large factors of safety is justified because the expense involved in preparing designs to approach exact requirements would far exceed the savings in weight and material.

The same economic considerations that justify overdesign in low-production-volume equipment demand designs of low weight and high stress in many machine parts where weight is all-important, as in airplanes or in large-production-volume machines, such as automobiles, where both weight and cost must be considered. Obviously, the dynamically loaded parts of such machines should be designed with accurate knowledge of their fatigue strength.

Laboratory Fatigue Data

When we try to apply quantitatively the accumulated laboratory fatigue data to such design problems, we find that they are almost useless. Published data on fatigue assume that:

1. The operating stress can be determined.
2. Laboratory test specimens are representative of a material when that material is formed into a machine part.
3. The amount and nature of the applied load is known.
4. Load variations occur in an orderly and predictable manner.
5. Representative fatigue curves can be constructed from a dozen or less specimens.
6. Machine parts must be stressed below the fatigue limit to be successful.

These assumptions are not justified in practical design.

Stress Cannot Be Calculated

From the data on internal stresses that have been discussed, we may reasonably have some misgivings about the reliability of our stress calculations. From experience with practical machine parts we can only conclude that stress calculations by textbook methods are wholly inadequate unless we generously temper our calculated results with experience. For example, by the usual methods of calculation, crankshafts may be stressed to 20,000 psi, connecting rods may be stressed to 40,000 psi, valve springs 90,000 psi, disc clutch springs to 180,000 psi, while another form of disc spring supports, by calculation, 600,000 psi. Obviously, some of these stress values are ridiculous, but the formulas used in each case conform to the "laws" of mechanics. The actual stress in crankshafts is probably several times 20,000 psi, while the 600,000 psi in the disc spring is not reached because of yielding in local highly stressed regions.

The unreliability of stress calculations has almost been forgotten by seasoned designers because they no longer take the numerical values of their stress calculations literally. Instead, they have learned by experience that, by the usual methods of calculation, the numerical values have different meanings for different machine parts; that is, somewhat rough empirical correction factors are applied.

Extensometer Readings of Doubtful Value

There is a growing interest in various devices employed to make direct measurements of stress, such as by photoelasticity, brittle lacquers, extensometers, and similar instrumentation, in the belief that these devices will provide accurate stress data. The accuracy of stress data from such measurements is usually greater than can be obtained from the most involved mathematical analysis, but that they are far from reliable can easily be shown by fatigue tests. Two specimens may vary widely in fatigue strength depending upon minute differences in surface finish or internal stresses. Since internal stresses are often desirable and are frequently unavoidable due to processing operations, such as machining, heat-treating, straightening, or grinding, as has been discussed, and since surface finishes vary all the way from rough forgings to lapped or honed surfaces, there is little reason to expect accuracy from extensometer readings, and even less for photoelastic tests, since photoelastic specimens must be free from internal stress and must be made of another material.

Photoelastic and extensometer readings are measures of elasticity in which the changes in dimensions are the statistical average of all of the material involved in the measurement. Fatigue tests provide a strength measure of the weakest portion of the material involved, usually at the surface, even though it be submicroscopic in size. Obviously, we cannot expect agreement between fatigue measures of stress and the stress readings obtained from elastic measurements alone.

Even if stress could be determined, the fatigue data from laboratory specimens could not be used because machine parts cannot be finished with the care and exactness that is given laboratory specimens. Abrupt section changes cannot be avoided, high internal stresses are often present as a result of processing or because of local heating as from bearing friction, surfaces are subject to bruises and to corrosion of various kinds. These effects cannot be evaluated in terms of arbitrary stress-raisers in controlled laboratory specimens.

▪ Operating Loads Rarely Known

In the kind of machines under discussion, the dynamic loads are rarely constant for any appreciable time but vary up and down the load scale in an unpredictable manner. Only a small percentage of the total number of stress cycles are at maximum load, and this percentage will not be the same in the hands of any two operators.

This brings up the question of damage by overstress and recovery by understress as has been observed by several investigators in tests of laboratory fatigue specimens. No doubt such effects occur also in dynamically loaded machine parts, but how are such laboratory data to be applied to machine parts when the schedule of overload and underload is beyond control?

The development of engineering materials, designs, and processes requires that we conduct laboratory tests by which these factors may be evaluated, but to devise a reliable laboratory test is far from simple. The common belief that we can reproduce the conditions of service in a laboratory test is wholly erroneous. By the time the laboratory investigator on any particular part has provided for all of the conditions that occur in service he will have a complete machine in actual service.

The useful strength of materials in dynamically loaded parts is the fatigue strength of such parts under actual service conditions. No other measure will suffice. The strength of a part cannot be determined by tests of the kind that are commonly made in laboratories, many of which are used because they are easy to perform and not because they give useful information.

▪ Compromise Treatments

Many materials and processes have been graded and are still being graded by laboratory tests which are now known to have been very costly to industry. For example, the fiction that a carburized part should have a hard case to resist wear, and a tough core to resist breakage, arose from laboratory impact tests. In these tests, the strength of the part was judged by the number or intensity of hammer blows it would withstand before fracture. Since gear teeth resisted impact fracture in accordance with the physical properties of the core, it seemed logical to specify heat-treatments to bring out the best compromise between the imagined requirements of the case and the core. Being compromises, these heat-treatments were not the best for either region.

If, instead of counting the number of impacts or measuring the intensity of hammer blows to produce fracture, the gear tooth had been examined after the first impact, the tooth would have been found bent, and therefore ruined, and it would make no difference how many more blows were required to fracture the tooth.

This compromise heat-treatment resulted in reducing the quality of many millions of gears before it was realized that gear teeth fail by fatigue and that fatigue failure, for the usual depth of carburizetion, always originates at the surface of the case. From this evidence, it became clear the heat-treatment should consider the requirements of the carburized case only, and that the properties of the core were relatively unimportant, because, in bending and in torsion, the core serves mainly as a stuffing for the case.

▪ Physical Tests

Several kinds of impact tests are still being used and impact specifications appear in many drawings, but no man can explain and substantiate the significance of the test in terms of the service strength of machine parts.

Elongation and reduction of area are carefully measured and are prominent in our specifications, but we do not know their meaning in terms of serviceability of machine parts. We are told that "brittleness" must be avoided, but no matter how brittleness is defined it does not explain why this property is necessarily more harmful than ductility. Most machine parts that are plastically deformed are just as surely failed as if they were broken. We are asked to believe that machine parts generally must possess relatively high ductility and they must therefore be heat-treated to develop this property. However, when we really get down to applying severe dynamic loads, we forget about ductility and specify high hardness that certainly is well within the range of "brittleness" in the usual meaning of the word. Strong fatigue-resistant gear teeth are file hard. Wristpins, ball bearings, roller bearings, shafts, and cams are hard, and they are strong and fatigue-resistant because they are hard.

A gear tooth is just as surely a spring as the coil that actuates a valve. Why, then, must the one be hard and "brittle" and the other be relatively soft and "ductile"? Why can we not avail ourselves of stronger hardened materials? The answer may lie in our concept of brittleness. We do not fear "brittleness" from hardness when hardness is obtained by nitriding. Nitrided surfaces are not notch-sensitive because they are stressed in compression.

Notch sensitivity is probably the inability of a non-ductile material to yield locally and thus reduce tension stresses in local highly stressed regions, such as notches and scratches. The amount of ductility that is required to overcome "brittleness" depends upon the amount of yield that is necessary to reduce local tension stresses. If the surface is sufficiently pre-stressed in compression, local yielding is not required and therefore non-ductile materials will not be "brittle." As we improve our understanding of "brittleness" we may expect to use steels at higher hardness in many parts for which we now specify ductility. We will then gain from the greater inherent strength as well as from the increased strength obtained by compressively stressed surfaces.

The most significant of our easily performed laboratory tests is hardness. Since the static strength of most materials is roughly proportional to hardness, we will know the approximate static strength of a part if the hardness is accurately measured. However, the popular hardness testers such as Brinell or Rockwell are incapable of the accuracy that is required because they penetrate too deeply into the material being tested, and therefore they do not measure the characteristics of that most important part of the material, the surface layer.

Chemical analysis can only indicate the responsiveness to heat-treatment and can measure the potential strength of a steel only by its probable hardenability. Since strength is proportional to hardness, all properly heat-treated steels of equal hardness are equally strong regardless of their compositions.

Laboratory hardenability tests are now coming into general use. This test has much merit provided that we understand its meaning and that we do not debase it, as we are so prone to do by applying arbitrary hardenability specifications without considering the requirements of each particular part. Through-hardenability (approximately uniform hardness through the section) can be very important

for parts that are stressed in tension, but it is difficult to see why through-hardness is necessary in parts that are loaded in bending or in torsion, because in such members the stress decreases somewhat linearly with depth, reaching zero at the neutral axis. For this kind of loading, it would seem to be more important to develop heat-treatments that give the type of internal stress shown in Fig. 15 because, being pre-stressed negatively to the applied tension load, the dynamic load-carrying capacity is greatly increased.

The standard laboratory tensile test is, of course, incapable of indicating the useful bending or torsion strength of pre-stressed specimens, particularly when the pre-stressing is deep, as shown in Fig. 15. For such specimens, the tensile test cannot even distinguish between harmful and beneficial pre-stressing. Both would probably show decreased tensile strength, whereas under dynamic bending or torsion loads one would show greatly decreased fatigue strength and the other greatly increased fatigue strength.

However, we have done a reasonably satisfactory job in the past without worrying overmuch about the shortcomings of the methods used. We may be certain that we will do better in the future as more experience is gained and it is in the accumulation and organization of this experience that we can best serve the needs of the future. It is probable that fatigue studies will play increasingly important parts in future designs; but these studies will be based on fatigue tests of actual, full-scale machine parts instead of on laboratory specimens.

■ Fatigue Tests on Machine Parts

Fatigue tests of full-scale machine parts have been made by many laboratories for a long time, but since these tests have usually been made for the purpose of comparing one material, design, or process with another material, design, or process, the tests have been run at arbitrary constant loads without thought to the fatigue-curve characteristics and often without adequate correlation with service requirements. Because of this procedure, we have made little use of the vast quantities of such fatigue data as are now locked in our files in so far as establishing a basis for evaluating material, design, or process for the future is concerned.

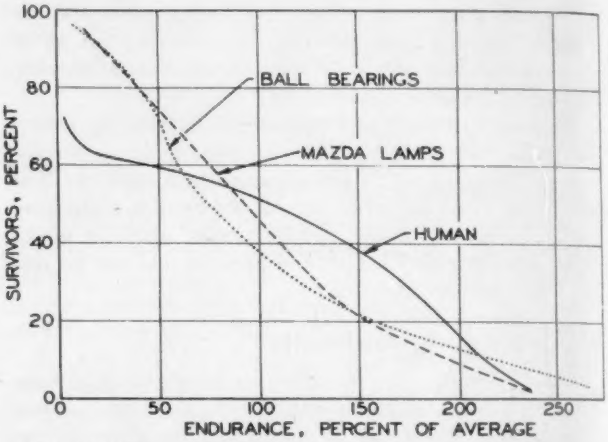
In the few cases where fatigue data on machine parts have been properly organized, we find that they reveal astonishing amounts of fundamental information about the many variables that are present in machine elements, many of which are not even qualitatively revealed by ideal laboratory fatigue specimens.

■ Fatigue Data Are Mortality Data

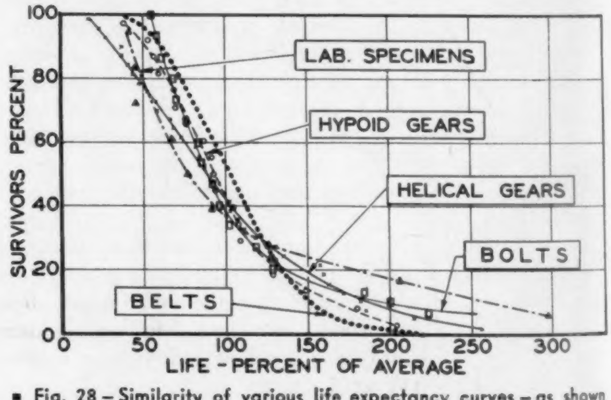
Fatigue data are mortality data and it is just as absurd to expect that reliable actuarial tables can be constructed from mortality data on a half-dozen individuals as to expect that reliable comparisons can be made from fatigue tests on a half-dozen machine parts. When a sufficient number of machine parts are fatigue tested at constant load and plotted in the manner of the well-known mortality curve for human life expectancy, we find remarkable similarity to human mortality experience. Heindlhofer and Sjovall[11] have shown life expectancy curves for commercially identical ball bearings, for commercially identical mazda lamps, and for human beings. These curves are shown in Fig. 27, in which the ordinate is the percentage of units surviving and the abscissa is durability in per cent of average life.

Fig. 28 is a life expectancy chart at constant load for commercially identical transmission gears in complete automobile transmissions, for commercially identical rear-axle gears in complete automobile rear axles, for commercially identical automobile fan belts[12], for commercially identical bolts, and for a group of ideal laboratory fatigue specimens. Similar life expectancy curves will result whether applied to mountain ranges or to the hairs on our heads.

Although the general form of all life expectancy curves is the same, they differ in detail. Note that the expectancy curves for machine parts (Figs. 27 and 28) do not extend to zero life as is the case in the human expectancy curve. Infant mortality is avoided in machine parts because the parts having a low potential life are rejected by factory inspection, a practice that is not followed for humans.



■ Fig. 27 – Comparative endurance – life expectancy curves for ball bearings and lamps compared with the life expectancy curve for human beings



■ Fig. 28 – Similarity of various life expectancy curves – as shown by machine parts, belts, and laboratory specimens

■ Variation in Durability

Another important difference is the relative life span for various machine parts. Note that for automobile rear-axle gears, the life span of the most durable unit was about four times the life span of the poorest unit, but for automobile transmission gears the life ratio from the best to the poorest was about 15 to 1; that is, childhood mortality is higher in automobile transmissions than in automobile rear axles. The life span ratios given should not

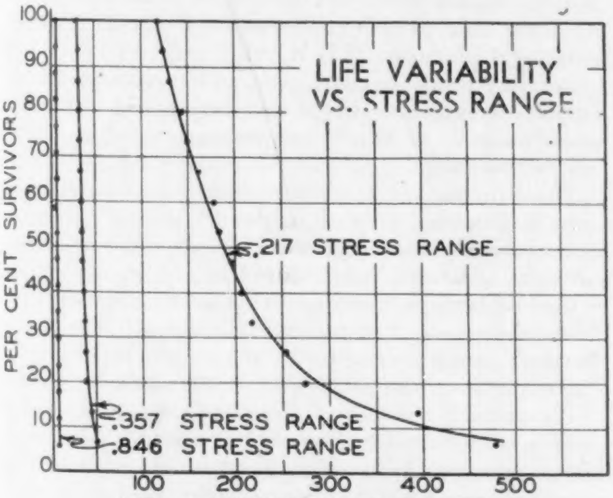
[11] See *Mechanical Engineering*, Vol. 45, October, 1923, pp. 579-581: "Endurance-Test Data and their Interpretation," by K. Heindlhofer and H. Sjovall.

[12] Private communication, R. S. Carter, Goodyear Tire & Rubber Co.

be taken literally because there are not enough test points in either curve to define their limits. As the number of test points is increased, the life ratio of the best to the poorest will increase but the "scatter" will be greater for transmission gears because the variability of stress resulting from end contact is greater.

The percentage variation in life of machine parts will also change as the test load or load range is changed. When tests are conducted at high load or high load range to produce fatigue failure after relatively few stress cycles, the percentage variation from the best to the poorest will be less than if the test is conducted at a lower load to produce fatigue failure after a relatively large number of stress cycles. This is shown in Fig. 29, in which the life variation of the bolts used to determine Fig. 23 is recorded. Note short life and the small variation in life for the bolts that were given low initial tightness (large stress range), and the greater life and relatively great variation in life for the bolts that were given high initial tightness (low stress range). The reason for this variable will become clear when we examine the form of scatter band of fatigue data from a sufficiently large number of fatigue tests.

In the class of light machines where weight must be conserved, it will probably never be possible to design mechanisms to withstand all the abuses that are encountered in service. If an airplane engine, for example, should be so sturdily designed that the shortest lived of each of its numerous parts was failure-proof under all the abusive conditions that may be experienced in service, the engine would be so heavy as to be impractical. As we learn how to increase the durability of each machine element, we will reduce but not eliminate failure hazards. Instead, progress will demand that we take advantage of such improvements by reducing the weight or by increasing the power output.



■ Fig. 29 – Life variability of bolts for different stress ranges

■ Insufficient Test Data

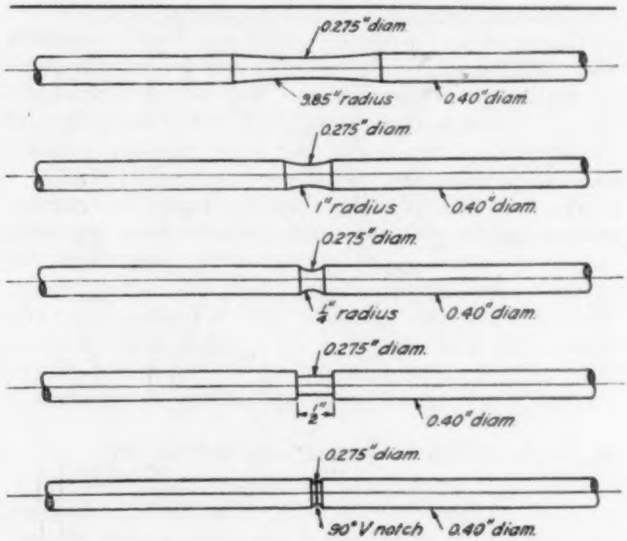
Reliable life comparison of machine parts demands a large number of tests unless the life difference is very great. It is obvious from the mortality charts that have been shown that, on the basis of a few tests, the poorer design, material, or process may rate higher than the better design, material, or process. Yet nowhere in the literature do we find fatigue data approaching even the minimum requirements of reliability. The reason is largely that most of the investigators in this field, particularly in work on steel, assume that we have no interest in data at any stress except the stress at which the specimen will endure indefinitely.

In practical fatigue testing of machine parts, it should be obvious that comparisons of material, design, or processes cannot be made unless the tests are run to failure and the comparisons are made on the number of stress cycles each will endure. This is true whether or not the part being tested is required to withstand, in service, a very large number of stress reversals at maximum load such as a crankshaft or a relatively small number of stress reversals at maximum load, such as chassis springs. Since all representative tests are made at loads that result in failure by fatigue, our interest lies not in the fatigue endurance limit where for steel, under most test conditions, life is infinite, but in that portion of the fatigue curve to the left of the "knee" where life is finite, that is, the sloping part of the curve.

■ Fatigue Curve Slope

The characteristics of the sloping part of the fatigue curve have been obscured in most of the published S-N plots (a) by never having enough test points and (b) by the popular custom of plotting fatigue data on semi-log charts. In the very few cases where published data contain a considerable number of test points, we find that when plots are constructed on log charts, the points tend to lie on a straight line instead of on a curved line as when they are plotted on semi-log charts.

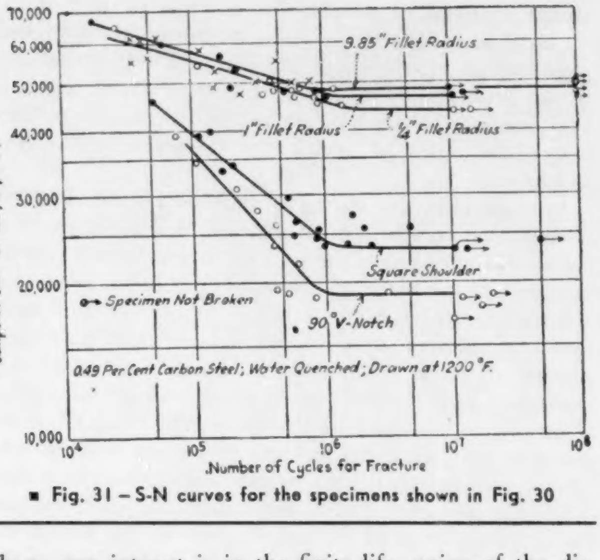
Fig. 30 shows a series of fatigue specimens used by Moore and Kommers[13] to determine the effect on fatigue of varying degrees of stress concentration.



■ Fig. 30 – Specimens used to study the effect of shape on endurance limit

The resulting fatigue curves, plotted on logarithmic coördinates, are shown in Fig. 31. The authors compare these specimens on the basis of calculated stress at the fatigue endurance limit; that is, the stress at the "knee" where the curve becomes horizontal. However, as stated

[13] See "Fatigue of Metals," by Moore and Kommers, McGraw-Hill Book Co.



■ Fig. 31 – S-N curves for the specimens shown in Fig. 30

above, our interest is in the finite life region of the diagram; that is, in the characteristics of the curve lying to the left of the "knee." Observe that as the "notch" severity of the specimen section is increased, the slope of the curve increases, and that the curves, if extended leftward, tend to cross one another.

■ Machine Parts Fatigue Curves

Fatigue curves of machine parts, no matter how well finished or how carefully rejected for detectable flaws, almost invariably show steeper slopes than are shown by well-finished fatigue specimens and, therefore, presumably the fatigue strength of a material as determined by ideal test specimens is not obtainable when that material is formed into a machine part. Permissible stress at the fatigue limit of a machine part may be less than 10% of the ultimate strength of the material, whereas laboratory test specimens may indicate 50% or more as obtainable.

The difference in slope of the fatigue curves suggests that this characteristic promises a way whereby we may eventually greatly improve our accuracy in determining the strength of machine parts. This is now being done in rating the load capacity of ball bearings, roller bearings, automobile transmissions, and rear-axle gears.

The lines plotted in Fig. 31 are intended to represent the averages for the specimens tested. Note the wide scatter of the test points and the increasing scatter of the points as the slope increases. Note also that, generally, the scatter decreases toward the left of the diagram. The significance of this scatter is not apparent in the diagram due to the limited number of test points, there being an average of only 12 failed tests for each type of specimen.

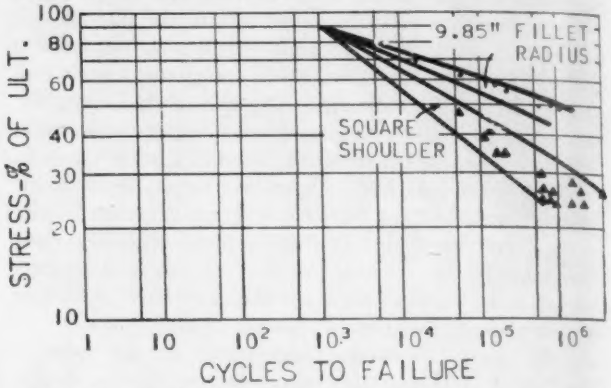
■ Slope Indicates Stress Concentration

The scatter of test points is due to unavoidable differences in test specimens no matter how carefully they are made. Since these differences constitute varying degrees of stress concentration, the fatigue line representing the poorest of a group of specimens should lie on a steeper slope than the fatigue line representing the best specimens. This is for the same reason that the average slopes of the specimens shown in Fig. 30 increase with the severity of the stress concentration as shown in Fig. 31.

[14] See Report of the Research Committee on the Fatigue of Metals, ASTM Meeting, June, 1941.

The test points for any group of specimens would, therefore, be expected to lie within a scatter band diverging from the region of high stress, and will be of the order shown in Fig. 32.

If a sufficient number of specimens had been tested, and if the stress scale proportionality were the same for all specimens, it is probable that the sloped lines in Fig. 31 would all tend to converge toward a point in the vicinity of 1000 cycles and 90,000 psi, somewhat as indicated in Fig. 32, in which the fatigue slopes of specimens 1 and 4



■ Fig. 32 – Probable form of scatter band of specimens 1 and 4, Fig. 30

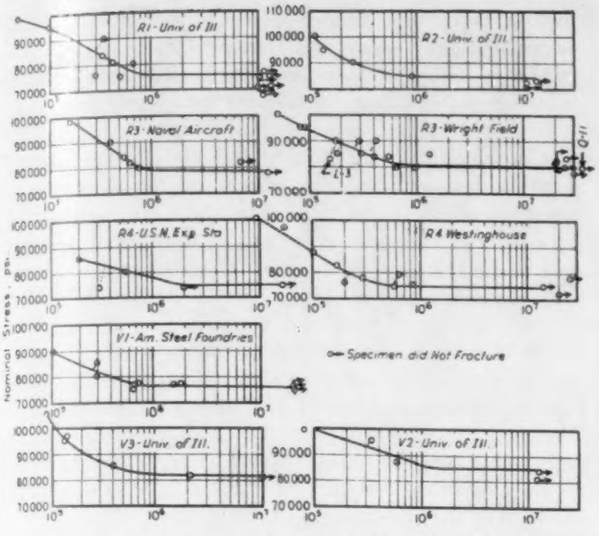
of Fig. 30 are shown as converging bands rather than lines. This region of intersection is suggested because the ultimate strength of the material tested by Moore and Kommers was approximately 95,000 psi and, obviously, if the stress scale is correct for each type of specimen, they would all have approximately the same strength at one stress cycle. The point of intersection would probably be at a considerable number of stress cycles because the ductility of the material permits adjustment of stress yield, thus reducing the influence of local highly stressed points.

For very brittle material, the intersection point of the fatigue curves for the type of specimens shown in Fig. 30 would probably be near the ultimate strength and nearer one cycle of stress.

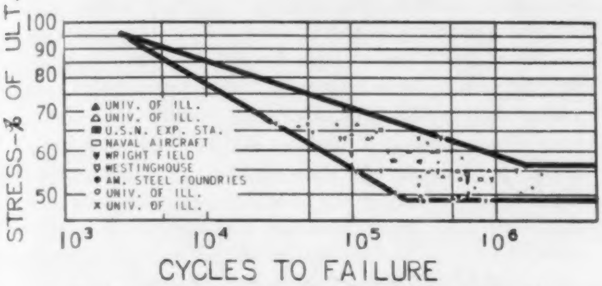
There are not now available sufficient data on any specimens to complete a group of fatigue diagrams to the region of intersection. Knowledge of the characteristics of fatigue curves at high stress would be valuable in industry since it would greatly facilitate interpretation of fatigue tests on machine parts. Such tests could be evaluated in terms of the slope of the fatigue curve, which would also give a clue to the actual stress, if desired, in the part being tested.

The research committee of the ASTM recently sponsored a cooperative test program in which several laboratories conducted independent fatigue tests on identical material (heat-treated SAE 4340) under similar test conditions. The results were reported in an ASTM research report[14] from which the group of plots shown in Fig. 33 were taken. Note the wide disagreement between the curves from the several laboratories in the fatigue limit as well as in the sloping part of the curves. When all of the 59 individual failed points are plotted on a log-log chart, as is shown in Fig. 34, we begin to see a semblance of order, in that all of the points lie within a scatter band of the same converging form as is shown in Fig. 32.

In passing, it is interesting to note that in Fig. 33 we find nine test points at 85,000 psi load, which are the points plotted in the life expectancy curve, Fig. 28, to show



■ Fig. 33 – Semi-log plot of data reported by the ASTM Research Committee on Fatigue of Metals



■ Fig. 34 – Log-log replot of data plotted in Fig. 33

that the life variation of laboratory specimens is of the same order as the life variation of gears and other machine parts.

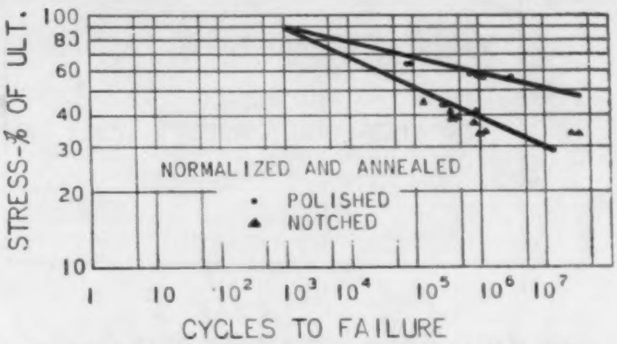
Published data on fatigue of metals contain numerous tests showing the same general trend of increasing slope with increasing stress concentration whether due to differences in specimen shape, specimen size, mechanical working of specimen surface, surface coatings, fillet radii, surface finish, or to variations between "identical" specimens. As previously stated, this tendency toward convergence is often not apparent in the published curves because the investigators have plotted their data on linear ordinates and logarithmic abscissa, and always there are insufficient test points. The following diagrams copied from published papers have, when necessary, been replotted for the sake of uniformity on logarithmic coordinates to the same scale as used by Moore and Kommers in which the stress scale is four times the scale of stress repetitions. The slopes of the curves are calculated as the measured horizontal distance multiplied by the scale ratio divided by the measured vertical distance: $\dfrac{\text{(Abscissa} \times \text{Scale Ratio)}}{\text{Ordinate}}$

Note that this is the reciprocal of the slope as ordinarily used in engineering, but it is a more convenient form.

[15] See ASTM Proceedings, Vol. 37, Part II, 1937, p. 199: "Fatigue Properties of Metals Used in Aircraft Construction at 3450 and 10,600 Cycles," by T. T. Oberg and J. B. Johnson.
[16] See *Modern Plastics*, Vol. 19, September, 1941, pp. 57-62, 78: "Mechanical Tests of Cellulose Acetate," by W. M. Findley.
[17] See *Luftfahrt-Forschung*, Vol. 18, March 29, 1941, pp. 102-106: "Über den Einfluss von Bohrungen mit Gewinden und Kerbverzahnungen auf die Zeit- und Dauerfestigkeit von Leichtmetall-Flachstüben," by H. Bürnheim.

Oberg and Johnson[15] report a comparison between polished and notched specimens, Fig. 35, with results similar to the exeperiments by Moore and Kommers, Fig. 31.

Surface treatment of the test specimens, other than the

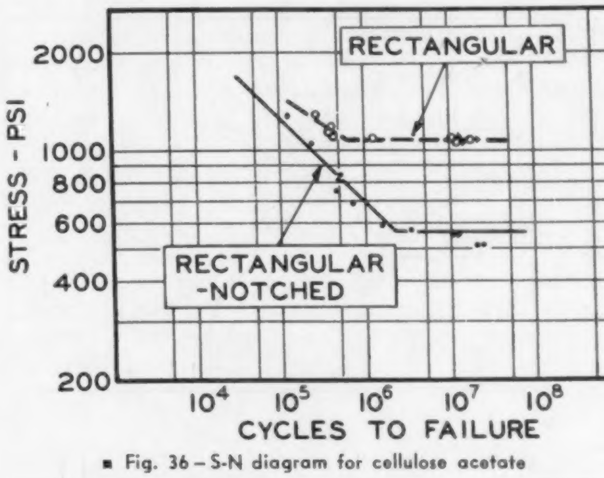


■ Fig. 35 – S-N diagram for a metal used in aircraft construction – 4134 steel

degree of smoothness, has a marked effect on fatigue strength. Horger and Maulbetsch[2] compared normal well-finished specimens with specimens that had been subjected to a rolling operation which introduced compressive stresses in the surface layer with the results shown in Fig. 1. Since the rolled specimens were pre-stressed in compression, the subsequent tension stresses during the test were reduced, as is shown in Fig. 6; hence the difference in the slope of the curves for the two types of specimens. Since this treatment would be ineffective in a tensile test, the lines should converge in the manner shown.

When replotted on log-log charts, published fatigue curves on other materials than steel exhibit the same tendency to converge toward the left and to increase their slope as notch effects are increased.

Findley[16] conducted fatigue tests on cellulose acetate specimens with the replotted results shown in Fig. 36. As

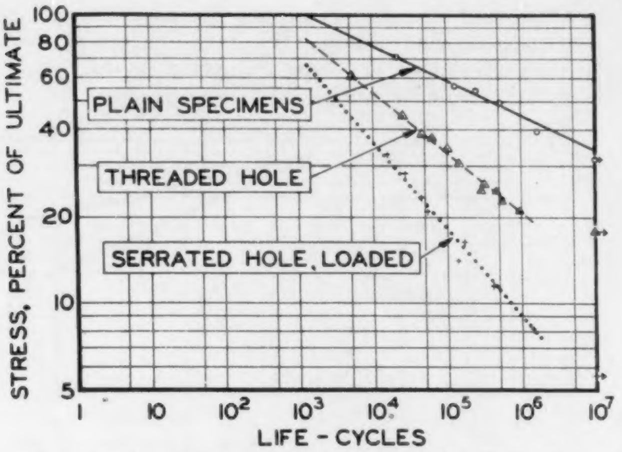


■ Fig. 36 – S-N diagram for cellulose acetate

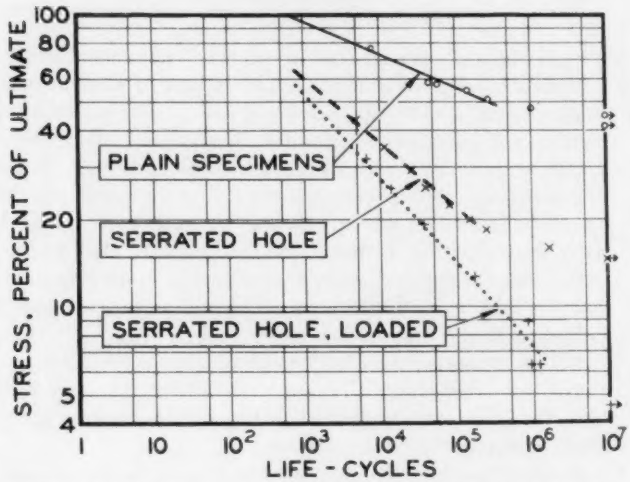
is usual, there are not as many test points as are needed to define the region of convergence but the trend is definite.

Bürnheim[17] reported fatigue tests on duralumin (Fig. 37), and magnesium (Fig. 38), in plain and various notched specimens. These replotted charts are more satisfactory than most published data in that the region of convergence is more clearly defined.

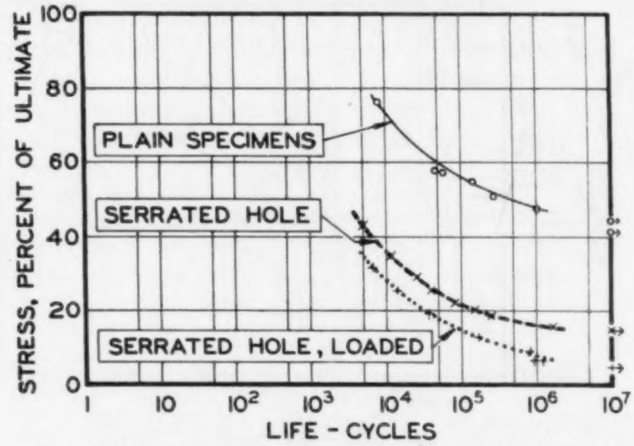
Fig. 39 is the original semi-log plot of the fatigue data on magnesium that is shown in the replot of Fig. 38. Note



■ Fig. 37 – S-N diagram for duralumin showing the effect of stress-raisers



■ Fig. 38 – S-N diagram for magnesium alloy (Electron) showing the effect of stress-raisers – log-log plot



■ Fig. 39 – S-N diagram for magnesium alloy (Electron) showing the effect of stress-raisers – semi-log plot

that when plotted in this manner, the data lose much of their significance and may not safely be extrapolated.

From the foregoing it seems reasonable, as a working hypothesis, to assume that, except possibly for very ductile metals, *the slope of the fatigue curve, as measured on a log-log plot, may be considered a measure of effective stress; and fatigue curves for varying stress concentrations converge toward a point near the tensile strength of the material and at some considerable number of stress cycles.*

It should be remembered that the tensile strength in a test in which the load is slowly increased is lower than in a tensile test in which the load is maintained for a very short time[18] as in a fatigue test, and also that there is a considerable variation in the tensile strength of any material as measured by a number of tensile test specimens. Therefore, the tensile strength on a fatigue chart would actually plot as a band and not as a line and would lie above the normal tensile value. Likewise, the lines of a fatigue plot would converge to a region above the normal tensile strength and would probably not meet at a point. However, the inclusion of these variables would considerably complicate the above hypothesis and since they occur in a region of the fatigue plot that has little or no practical value, they may, for the present be ignored.

The application of this hypothesis to the fatigue strength of machine parts has some important implications. A large variety of machine elements are constantly being tested for relative durability in the laboratories of industries engaged in the manufacture of light-weight, high-output machines. In most cases these fatigue tests are intended to compare one design, material, or process with another.

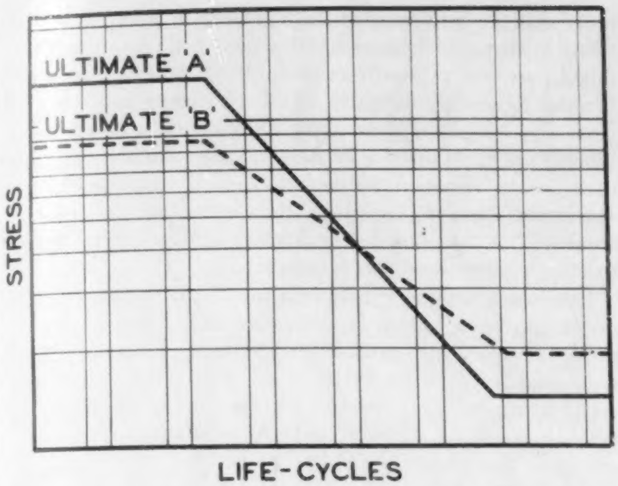
It is axiomatic that nothing can be learned in regard to limiting loads except through tests to destruction and, therefore, the fatigue tests for practically all parts are run to failure and the comparison is made on the number of stress cycles at constant load that each part will withstand. As stated above, this procedure is followed regardless of whether, in practice, the part in question is stressed below the fatigue limit or whether it is a part requiring relatively short life at maximum stress.

This method of evaluating test results is subject to serious error for several reasons. If it is true that fatigue curves radiate from a point in the high-stress region, it is obvious that comparisons of specimens cannot be made on a percentage basis only, since the percentage difference will vary all the way from zero to infinity depending upon the load that is applied during the test. Furthermore, since the scatter band for each test part should also radiate from the same point, as was shown in Fig. 32, the width of the band in terms of life may be several hundred per cent, as shown in Fig. 34; therefore, unless a considerable number of tests are run for each part, there is no assurance that whatever life difference is found is not just the chance location of these particular test points within the scatter band. It is easily possible that the better design, material, or process will apparently rate lower than the poorer design, material, or process if insufficient tests are made.

It is possible that the average fatigue curves for two materials having different tensile strengths and yield points will cross at some point in the finite life region due to differences in sensitivity to stress-raisers. In such cases, life comparisons may be positive for one material at one test load and negative for the same material at another test load. The diagram shown in Fig. 40 illustrates such a situation.

It is evident, therefore, that true comparisons can only be obtained through fatigue tests on a sufficient number of parts at varying loads to outline the slopes of the scatter-band limits. While this may appear to be an impractical requirement, it is not so difficult as it seems. It is only necessary that the results of the present routine tests be

[18] See Institution of Mechanical Engineers Proceedings, Vol. 141, April, 1939, pp. 175-185: "Deformation and Fracture of Mild Steel under Cyclic Stresses in Relation to Crystalline Structure," by H. J. Gough and W. A. Wood.

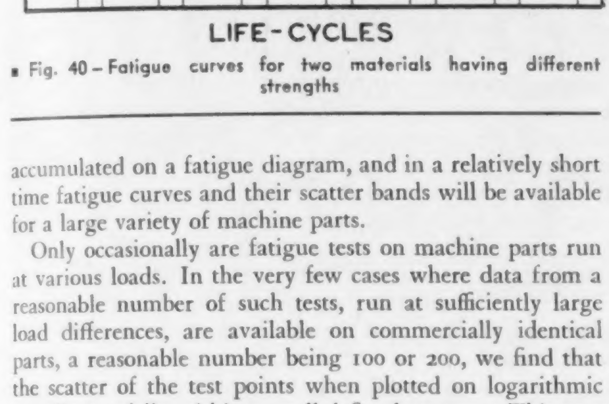


▪ Fig. 40 – Fatigue curves for two materials having different strengths

accumulated on a fatigue diagram, and in a relatively short time fatigue curves and their scatter bands will be available for a large variety of machine parts.

Only occasionally are fatigue tests on machine parts run at various loads. In the very few cases where data from a reasonable number of such tests, run at sufficiently large load differences, are available on commercially identical parts, a reasonable number being 100 or 200, we find that the scatter of the test points when plotted on logarithmic coordinates falls within a well-defined pattern. This pattern tends to radiate from a point at high stress and a low number of stress cycles and to diverge to a broad band at low stress and high number of stress cycles just as was suggested in Fig. 32.

This is clearly shown in Fig. 41, which is a fatigue diagram of about 200 complete automobile and truck rear axles of various makes and sizes.[10] The stress scale shown in this diagram is not actual stress but is believed to be



▪ Fig. 41 – Fatigue diagram showing scatter band for fatigue tests of 196 spiral bevel rear-axle gears

proportional to actual stress. The axles were tested at loads to produce failure of one or more pinion teeth through the range of from 7000 cycles to 1,000,000 cycles.

The scatter of the test points is due to variations in one or more of the many variables that are always present in commercially similar parts, such as internal stresses, fillet radii, cutter scratches, bearing, shaft and housing deflections, warpage in heat-treatment, and so on.

The slope of the average durability line, calculated as the horizontal distance divided by the vertical distance measured on logarithmic coordinates is approximately seven, while the slopes of the upper and lower limits of the scatter band are respectively nine and five.

The intersection point at the left of the diagram should lie near the ultimate strength of the material (approximately 300,000 psi) which, if proved, would supply us with a measure of actual stress for the entire diagram.

The diagram shown in Fig. 41 is not ideal as a proof ot the scatter band or the intersection point, since it includes a variety of axles made from various alloy steels variously heat-treated, for which the stresses were calculated by an empirical formula.

Satisfactory determination of the characteristics of the scatter band would require a large number of fatigue tests on one form and size of specimen made of one type of material similarly heat-treated and tested to produce failure over a range of stress repetitions from as near a single cycle of stress as possible to the fatigue limit.

Data approaching these requirements[19] have been accumulated by the various ball- and roller-bearing manufacturers, but the published data are not yet extensive enough to define the form of the scatter band. Particularly, more data are needed in the very low stress range and in the very high stress range.

However, fatigue data on ball and roller bearings need not in all particulars agree with fatigue data on other forms of machine parts, since failure of rolling bearings usually originates below the surface of the material. Surface influences, which play so important a part in fatigue of ordinary machine parts are, therefore, absent in rolling bearings. This would be expected to influence the permissible stress, and possibly the form of the scatter band. The scatter band as reported by Macauley[20] and by Styri[21] is parallel to the average life curve throughout the life range shown.

Ball and roller bearings are also peculiar in that their curves do not show a fatigue endurance limit as is usually found in fatigue specimens. According to the catalog ratings, the sloped lines continue to more than a billion inner race revolutions, and since there are several stress cycles per revolution, we do not find a "knee" in these curves up to more than five billion stress cycles.[22]

We seek to determine actual stress only as a step in predicting the adequacy or inadequacy of our designs. Any other means that will enable us to predict the performance of our designs will do as well. Ball- and roller-bearing manufacturers do not consider stress at all in their catalog ratings but rely entirely upon tabulated load capacities as determined by service experience that has been correlated with laboratory test data on complete bearings. In practice, we are not only unable to calculate or to measure but we do not even know the manner of load applications in service on the majority of machine parts.

Laboratory fatigue testing of light-weight, high-output machine parts, as well as other laboratory tests such as on fuels, oils, tire wear, and so on must be definitely correlated with service data on the part in question before the results can be accepted. This requires that, for fatigue, tests must be devised that will agree with failures that occur in normal service as to the location of points of fracture and the character of the fractures. It is not important that the test procedures agree with the preconceived notions of service loading.

[19] See *The Ball Bearing Journal,* No. 3, 1927, SKF Industries, Inc.
[20] See *The Automobile Engineer,* Vol. 13, July, 1923, pp. 213-223: "The Endurance of Ball Bearings," by A. W. Macauley.
[21] See *Mechanical Engineering,* Vol. 47, June, 1925, pp. 490-492: "General Properties of Ball Bearings," by Haakon Styri.
[22] See *The Ball Bearing Journal,* No. 3, 1937, SKF Industries, Inc.

The slope of the finite life portions of the S-N diagram has been discussed from the standpoint of fatigue tests at constant stress range. Most of the test data presented hereto have been taken from the specimens in which the stress was completely reversed. However, many machine parts are otherwise stressed, as for example, the case of properly tightened bolts in which the stress range approaches zero. Valve springs are stressed through a relatively narrow range in one direction only, being preloaded to approximately 25,000 psi stress, which is increased to approximately 90,000 psi when the valve is fully open. Gear teeth usually are loaded from zero to a maximum stress in one direction only. Crankshaft stresses may be somewhat more complex, being completely reversed in bending during each revolution, while transmitting torque in one direction only.

Many experiments have been conducted to determine the effect of varying the stress range, but again interest lay in the stress at the fatigue limit, and few data are available on the change of slope of the curve with stress range. However, since the stress at the fatigue limit increases as the stress range decreases, as has been amply demonstrated, it follows that the slope must decrease (become flatter) as the stress range decreases.

Moore and Kommers[13] present a modified Goodman diagram from which the fatigue slope for ideal specimens

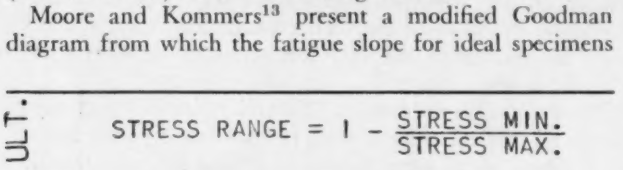


Fig. 42 – Effect of stress range on fatigue life – the slope of the fatigue curves varies with the stress range as well as with the stress concentration

may be constructed for any stress range. Fig. 42 shows a replot of this modified Goodman diagram for three stress ranges. The upper curve represents a small stress range similar to that of automobile valve springs; the second curve represents a stress range of from zero to maximum as for gears; and the lowest curve represents complete reversal of stress as may occur in a crankshaft.

The slopes of these curves are respectively 80, 48, and 17. We thus see that the slope of the fatigue curve varies with the stress range as well as with the stress concentration and, therefore, the hypothesis that the slope of the fatigue curve is a function of the usual conception of stress is no longer tenable. If, however, we state that the slope of the fatigue curve is a function of effective stress, the hypothesis will apply for any stress range.

The methods now used for coordinating laboratory tests with service experience are too haphazard to be completely reliable. Service failures must, obviously, be infrequent, and when true fatigue failures do occur, it is usually the result of harder-than-normal service combined with a specimen lying on the lower fringe of the fatigue scatter band.

Since failures must be infrequent, it is highly important that failed parts be examined by competent observers in order that the true cause of the trouble may be determined. Clear evidence of fatigue failures does not prove that the failed part was primarily responsible. A bolt may fatigue because it was not properly tightened during assembly; a gear may fatigue due to improper support or to a failed bearing; a crankshaft may fatigue due to inadequate or maladjusted vibration damper; and so on without end. It sometimes happens, therefore, that immediate corrections are made to the wrong part and recognition of the true trouble is sometimes greatly delayed.

Laboratory fatigue tests on machine parts must not only duplicate service failure as to location of fracture, but they must, in some cases, produce failure in approximately the same number of stress cycles if accurate life comparisons are to be made. This requires that we distinguish between normal operating stresses and the relatively infrequent overloads that cause the failure.

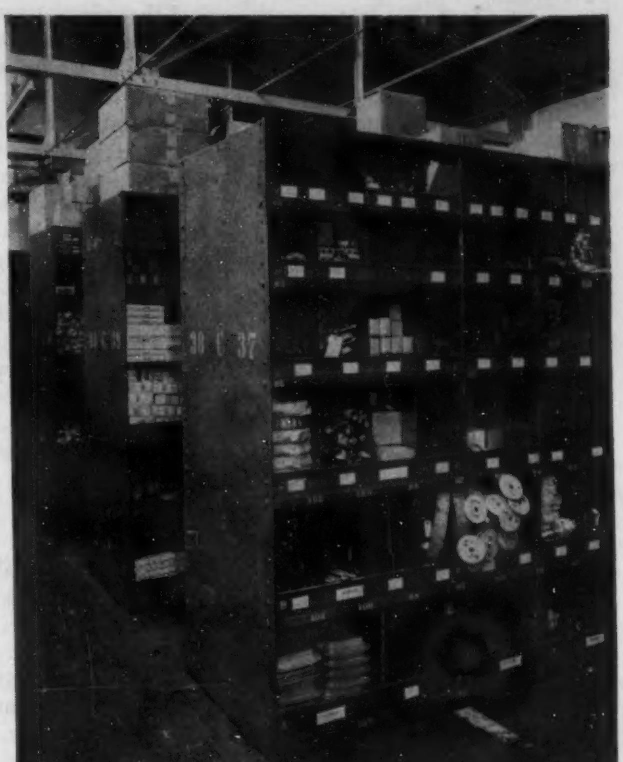
Fully 90% of all fatigue failures occurring in service or during laboratory and road tests are traceable to design and production defects, and only the remaining 10% are primarily the responsibility of the metallurgist as defects in material, material specification, or heat-treatment. While this ratio is not a measure of the quality of workmanship contributed by each department, there can be no doubt that the metallurgist has a better appreciation of his responsibility for fatigue failures than has the designer, the engineer, or the man in the production department.

You are familiar with the routine that is followed when a failed part is received by the laboratory. The fracture is examined and is found to be due to fatigue, the material is analyzed for composition, sections are studied for all the many things that are metallurgically important and a report is written describing the things that are and are not up to par. But no matter how many possible metallurgical causes of trouble are found, such examination is far from sufficient unless the failure is also examined for design faults and possibly bad lubrication and assembly practice. Most of the failed parts should not be sent to the metallurgist at all, but unfortunately very few engineers or production men are adequately trained in diagnosing fatigue trouble and, therefore, failures are seldom examined for contributing mechanical causes. Most of our engineers pass all fatigue problems on to the metallurgical department with the implication that something must be wrong with the material or with the heat-treatment.

The study of fatigue of materials is properly the joint duty of the metallurgical, engineering, and production departments. Unless all of these departments have an understanding of fatigue phenomena and the factors that promote fatigue, they cannot recognize their individual responsibilities for the product they manufacture. There is no definite line of demarcation between mechanical and metallurgical factors that contribute to fatigue, and there must, therefore, be very close cooperation between the metallurgist and the engineering fatigue specialist, if such there is, or the metallurgist must possess the qualifications of the metallurgist, designer, and machinist. This overlapping of responsibility is not sufficiently understood in industry and hence the engineers are constantly demanding new metallurgical miracles, instead of correcting their own faults. It would be very helpful if the metallurgists would be less willing to look for metallurgical causes of fatigue and insist that equally competent examination for mechanical causes be made. Until this is done, we cannot hope to make full use of our engineering materials.

WARTIME REPLACEMENT PARTS

by ROBERT CASS
Chief Engineer, White Motor Co.



IT has not been possible actually to canvass every manufacturer of motor trucks with an idea of finding out the degree of substitution that has been made in trucks built during 1941 and 1942, but it is believed the general comments as well as the detailed analysis in this paper will represent a fairly accurate picture of what has happened in this respect. In some cases, the timing or the date of such substitutions may vary considerably due to the inventory situation and other factors entering into truck production during the period under discussion. Later on in this paper, it is intended to analyze component parts of engines, clutches, transmissions, axles, and other functional parts in such detail as may prove helpful in preventive maintenance and in determining when replacement may reasonably be considered necessary. It can be stated, however, that the overall picture from the operator's standpoint is good. It is good in the sense that no substitutions have been made that it is believed will seriously affect the economy of operation to a marked degree; and by education of existing maintenance crews and drivers, and a closer contact with the manufacturer's service facilities, very satisfactory results should be obtained.

One of the important aspects of the situation is that to a great degree the handwriting on the wall with regard to substitution of particular materials was seen early enough to enable some studies to be made, and in certain cases the time allowed has permitted accelerated tests. These studies and tests have been valuable in preventing the overnight changes which we all know can be somewhat disastrous.

It should also be made quite clear at this stage that commercial trucks which are now being currently built under orders of the War Production Board are not such that

[This paper was presented at the SAE National Transportation and Maintenance Meeting, New York City, May 6, 1943.]

WARTIME shortages are making "conservation" the watchword of the average citizen as well as of the operator of large fleets of motor trucks. Since the fleet operator has always practiced a certain amount of conservation, he is apt to feel that he is already doing about all he can to protect his machines.

In this paper, Mr. Cass shows why it is necessary for the truck operator to reappraise his situation, not overlooking any possible method whereby parts can be made to last longer. He also suggests a program for carrying out this fundamental purpose of taking care of what we have:

1. Study particularly all parts being used that include less satisfactory substitutes.

2. Study overloading to see if it can be reduced or eliminated entirely. Make it a rule not to exceed recommended engineering ratings, except under unusual conditions.

3. Cut down engine speeds.

4. Educate each driver to his part in the conservation program.

As the present equipment gets older and older, Mr. Cass stresses, it is essential that there be no let-up in the practice of this preventive maintenance – rather it will be necessary to buckle down and practice it more faithfully than ever.

■ ■ ■

THE AUTHOR: ROBERT CASS (M '39) graduated from Crayford School in 1912 and – after an interruption of his studies due to the war – from London University in 1919. He majored in engineering in college, and has devoted his enthusiasm and energy to engineering problems ever since. Chief petty officer in the Royal Naval Air Service from 1915 to 1918, his first position after the war was with Short Bros., aircraft manufacturers of Rochester, Kent, England. He remained there from 1919 to 1924, when he accepted a position in America as an engineering instructor at Harvard University. After a year of teaching Mr. Cass joined the White Motor Co. as an engineer and continued as such until 1938, when he was made assistant chief engineer. He was promoted to chief engineer in 1941.

operators should be apprehensive about their overall performance. In other words, as we see it for the moment, they are not going to get any ersatz trucks. The problem of adequate and correct maintenance of motor transportation should and must be a matter of great concern to everyone. This is particularly true not only from the economics of the situation but because of the critical situation with respect to new trucks for replacement purposes.

There are approximately 4,800,000 trucks in service at this time, and in a normal year there are at least 400,000 to 500,000 trucks built for replacement purposes. During the past year, with rationing of trucks by ODT, there were 43,130 trucks delivered to operators for replacement and expansion. Allowing for the light-truck portion of the 500,000, these figures clearly indicate the fact that trucks are wearing out at a much faster rate than they are being replaced. In addition to this serious situation, motor trucks are being called upon to do a much greater amount of work than they are normally subjected to. Obviously, this comes about in several different ways. One is the heavy loads of material being handled, and another is the greater number of hours that power equipment is being worked in order to take care of constantly increasing traffic that the war effort calls upon trucks to handle.

■ Critical Situation

I think we have to face the fact that this is a very critical situation and that it could very well, somewhere along the line, break down to a very serious degree. This situation is further complicated by the depletion of trained drivers and maintenance men. Where operators are doing their own work, it becomes of supreme importance to see what educational means can be instituted to slow down the rate of deterioration. Maintenance, therefore, instead of becoming more effective, can easily become less effective; and this is in a time when for purposes of conservation of the vehicles the best is none too good.

The parts situation is also critical, so that this vicious circle is completed by an increasing demand for parts to take care of inadequate maintenance at a time when the number of parts, of necessity, is limited. Later on in this paper it is proposed to restate some of the fundamentals and offer as a guide suggestions with regard to their observance. The powerplant is obviously the one part of a truck which must necessarily be given the greatest possible care in these times, representing as it does 80% of the maintenance cost.

Before we start to analyze in detail the units, it is obviously fair to state at this stage that the performance of the substitutions in any vehicles cannot yet be confirmed by actual field experience. It will be realized that, in many cases, the substitutions have been of recent date, and we can only offer suggestions anticipating probable trends.

In attempting to predict the probable life of substituted parts, I think I should first offer a definition of the word "prediction." It has been said that, in terms of today's uncertainties and naturally indefinite conclusions, a prediction can be justifiably defined as: "A 100 to 1 guess on an unknown subject – supported by incomplete and inaccurate information obtained from unreliable sources, and presented by some party incapable of interpreting it even if it *had* been of any value in the first place."

With respect to National Emergency (NE) steels, these were adopted overnight and have not had the benefit of operating experience. Under normal conditions, a new steel would require at least two years of tests before it could be adopted. These steels have had concentrated tests by the best metallurgical laboratories in the country, but the information obtained is so varied that it might not be readily applied to a particular job in time to realize its fullest use. For example two manufacturers' laboratories reported favorably on a new steel for axle shafts; the third could not use it because their axle shaft was much larger and did not respond to heat treatment so readily.

The new steels developed for the emergency are the 8000 and 9000 series. The 1300 series being used by the War Department is not new as it has been in use for 30 years. The 8000 series was developed last year and was finally adopted after being subjected to intensive laboratory tests. With one exception, they were abandoned, however, because the series contained a small amount of alloys, notably molybdenum, which suddenly became too critical to use, and the 9000 series with a lower percentage of nickel, chrome, and molybdenum was adopted instead. These elements are contained in normal scrap and are not added as new materials. All three of these steels owe the greatest part of their strength to a large percentage of manganese, which is very effective in improving the strength of steel; in fact, better than some more generally accepted alloys. In the past, manganese steels have not received as much intensive development as they might have, it having been thought that manganese would be a critical war alloy. This has not occurred; therefore, an extended use is being made of manganese as an alloying material. Our lack of experience, as previously stated, makes definite recommendations difficult, but the following items should definitely be taken as recommendations and rigidly adhered to as far as possible:

1. Do not put sudden strains on axle shafts, which might start an incipient crack, which in turn would start a fatigue failure. Fatigue is the property which is still unknown with respect to these steels.

2. Do not clash gears, as the new steels are apparently more brittle when case-hardened than the old ones. Until we develop the technique of overcoming this brittleness, we may have some trouble.

3. Keep rigid inspection of all vital steering parts, and in particular, do not operate after there is lost motion in knuckle pins and other steering connections.

4. Do not use excessive engine speeds in going down hill with the engine used as a brake.

■ Pistons

Originally, pistons were made of primary aluminum, now they are made from secondary aluminum. It is particularly important to watch the warm-up period and, of course, as will be true in all these recommendations, use a good quality oil.

■ Cylinder Blocks

No changes have been made in acceptable standards for cylinder-block material. While it is impossible to guarantee that amounts of chrome and nickel can always be as high as desirable, the general experience seems to indicate that good foundries have enough residual alloy in the scrap that comes from risers, and so forth, to throw in the cupola and secure a good high Brinell iron. This insistence on a high Brinell is important, obviously, for many reasons, but

is of extreme importance when valve inserts are used and strength becomes of consequence.

Crankshafts

Carbon steels with a slight addition of chrome had been generally used for some years, before the emergency, for modern truck engines. This material is necessary to ensure good Tocco hardening. Here again, the residual alloy used in the steel companies' scrap seems to be adequate at this time to secure the requisite strength and hardness of shaft. The Tocco process is a good guide as to whether material is satisfactory, as minute surface cracks will develop where the combination is inadequate. Where vibration dampers containing rubber are used, more than usual care must now be exercised to keep oil from penetrating into the crude rubber. Natural rubber is employed in these dampers and reclaimed rubber cannot be used. Therefore it is very important to keep the dampers clean and tight on the shaft.

Valves

Naturally, some changes have had to be made in valve materials because of the high percentage of alloy used. Austenitic steel was currently used before the emergency in all parts of exhaust valves of hard-working truck engines. The industry is now using two-piece valves in which the amount of austenitic steel has been reduced. The question of valve clearances therefore becomes of great importance, as the coefficient of expansion changes with elimination of austenitic steel, and calls for a regular checking. Another very important item today with respect to valves is the mixture ratio. With the use of materials not as high as formerly in heat-resisting properties, a lean mixture must at all costs be avoided. From the point of view of life, a mixture slightly on the rich side is preferable. A check must also be made of the valve compartment from time to time to be certain that varnish build-up will not result in sticking valves. A sticky valve can easily mean a ruined valve; and it is worth the trouble of periodic inspection to prevent this type of failure. Storage conditions are just as important as ever: hot engines must not suddenly be put into storage conditions which are very cold and of high humidity. Reduction in the chrome can definitely accelerate corrosion, commonly known as "cold corrosion."

Valve Springs

Valve springs in the past have to a large degree been made of Swedish wire. Since such wire is no longer available, wire of the domestic variety has taken its place. It is interesting to note that, if anything, the domestic wire has stood up better than Swedish wire in laboratory tests. Here again it is very important that the conditions surrounding the valve chambers be such that they are fully protected against corrosion, as such corrosion very readily promotes valve spring breakage. Cleanliness is still next to godliness even in trucks.

Bearings

Where copper-lead bearings have been used in high-output engines, no change has been made. In engines using tin-base babbitts, the scarcity of tin has made necessary the use of lead-base babbitts. Tests indicate up to date that very satisfactory performance will be secured from the lead-base babbitts provided that the temperature of the oil does not rise to too high a level. The maximum allowable temperature is in the neighborhood of 225 F. The strength of the bearings is affected materially as the temperature is markedly increased above this maximum. Generally speaking, any bronze bushings that have been used in any part of the vehicle have been changed to steel-backed bushings coated with the bearing material. This procedure is not desirable in normal times as such a bearing is much more expensive and offers no advantage; in fact, in certain applications, the rusting of the steel is a definite disadvantage. Typical examples of substitutions in the engines are the wristpin bushings and camshaft bushings. A further possible disadvantage, which is not regarded as being very serious, is that the bearing temperature might be slightly raised, as the thermal conductivity of the bearing with steel backing is not quite as good as that of the plain bronze bushing.

Connecting Rods

No appreciable amount of substitution has been made in connecting rods, but driver care is very important in controlling engine overrun. Connecting-rod bolts are now of NE steels, and adequate safety is obtained under normal conditions.

Camshafts

Previously used materials and methods of heat-treatment of camshafts have not been changed.

Rubber Hoses

Up to date, rubber hoses from reclaimed rubber cannot be regarded as satisfactory. Tests are still being made, but generally speaking, the breakdown of the hoses is comparatively rapid and therefore should call for much more frequent inspection.

Water Pumps

Wherever possible, zinc- or chrome-plated parts of water pumps have been substituted for brass. Typical examples are the water-pump impeller and the water-pump gland nuts. Whereas in the past, brass tubing has been used throughout the cooling system, carbon-steel Parkerized tubing is now being used. In some sections of the country, the life of these water connecting tubes will be much shorter than others due to the difference in water used. Generally speaking, they are regarded as satisfactory only for the emergency.

Engine Gaskets

Soft carbon steel has been substituted for soft copper in engine gaskets, but the steel gaskets do not have the compatibility of copper. At the present time, however, we are being furnished with some gaskets with copper grommets in the water holes, but this is by no means certain to continue. Check testing with torque wrenches on the cylinder head is advisable. The manufacturer should be contacted through his service departments as to the proper torque wrench readings for steel gaskets.

■ Fan Belts

Here again a large portion of reclaimed rubber is being used in fan belts. Naturally, the mileage of these fan belts will not compare with those manufactured previous to the emergency, and they will call for more frequent inspection with respect to wear and tension.

■ Engine Thermostats

No radical substitutions have been made.

■ Carburetors

Many changes have been made in carburetors, mainly protective coatings of steel parts previously made from brass. Generally speaking, no coated part can be guaranteed, as minute parts of the surface may remain uncoated and the spot such a size as not to be revealed by normal visual inspection. In the case of carburetor floats, one may expect, every now and again, to find a "leaker" due to lack of complete covering. All gasoline lines have been changed over to steel tubing copper plated on the inside. The steel tubing generally used has a disadvantage in not permitting ready repairs in the field, as it is necessary to have special tools to handle such tubing. Flaring operations which are easily effected with copper tubing present a problem with steel tubing. Service stations of the truck companies and dealers, in many instances, can be helpful in this respect.

■ Ignition

The change from copper to stainless steel has not been too satisfactory, not because of the electrical requirements but because of the mechanical requirements of the more generally used fittings, which rely on wrapping effect of the core wire for good contact. Stainless steel, an extremely hard-surfaced material, calls for special care in seeing that fittings are firmly attached. Generators and starting motors are, to all intents and purposes, unchanged. In general, truck engines have been left pretty much intact, and where substitutions have been made, mechanical reliability has not been affected. However, under very hard working conditions, a shorter life may be expected.

■ Radiators

In some cases, it has been found practical and possible to substitute steel for copper fins in the radiators, retaining, however, the original copper tubes and brass tanks. This change did effect a marked saving in the amount of copper used but possibly introduces the need for a little more careful examination of radiators. The number of fins per inch has been greatly increased, making it easier for radiators to become clogged and raising the difficulty of assuring that no corrosion will take place on these fins. Operators, therefore, should be very conscious of inspecting radiator cores and keeping them as clean as possible. Moist particles left on fins are inevitably a source of corrosion.

■ Clutches

The principal change in the clutch deals with the lining. Where natural rubber compounds have been used in the past in linings, molded linings are now being furnished without any rubber fillers or binders. They should be generally satisfactory if the driver understands the necessity of care.

■ Transmissions

It is in transmissions that some of the greatest changes in connection with the use of emergency steels have taken place, and I think that because of the lack of experience as to the effectiveness of heat-treatments, we may here face a definite reduction in life. This reduction may be as much as 25% in the wear life of the respective gears. Furthermore, increased care is needed in shifting from one speed to another as it appears that substitute gears chip more readily when gears are clashed. Because of the generally very satisfactory transmission life secured in trucks in the past, the reductions in life which may occur may not prove to be too serious. Care and good oil are essential.

Note: In all these discussions it will be self-evident that cost cannot be considered, as money has ceased to have any value in wartime. Material is the determining factor in truck transportation today.

As far as the bearings are concerned, whether of the tapered-roller type or of the ball-bearing type, no changes have been made which seriously affect life, and tests conducted under comparable conditions bear out the favorable results in this respect.

■ Axles

The use of NE steels has been most significant in rear axles, though in some designs of axles their use cannot be realized without redesign. This would be true both from a strength as well as a life standpoint. In the designs in which substitutions of NE steels have been carried out, time alone will tell. As stated earlier, a good idea of how successful they can be will depend on the use of good lubricant at the proper level and the degree to which the driver will protect the rear axle by handling the starting and stopping of his loads.

■ Batteries

The battery situation has not changed in that, generally speaking, good batteries can be secured with composition cases and wooden separators and be perfectly adequate if properly maintained. Rubber threaded separators are splendid but not imperative. There is one thing, however, that operators should remember: Advance buying and storage at their shops of large quantities of batteries against a possible shortage is uneconomical and will result in very unsatisfactory conditions. Not only will the life of the battery be affected, but it will tend to create a universal shortage that no economics on the part of the operators could justify. Hoarding a product that deteriorates is not good business.

■ Steering

No sacrifice of reliability from a strength standpoint has been made in steering gears, but because of the use of emergency steels, again the wear life will possibly be less. Under such circumstances, examinations of steering gears should take place at more frequent intervals than has been usual in the past, and any undue play should call for the replacement of the part. Steering wheels themselves have

been changed from hard rubber to plastic material, but here again reliability has in no way been sacrificed.

I am indebted to the Bendix Products Division of the Bendix Aviation Corp. for a very detailed summary of substitutions, and while it is not intended to quote them in detail, wherever operators' problems exist the subject will be stated in full. Where cast iron has replaced brass, one may reasonably expect some rust formation, and therefore replacement will be necessary after shorter service than with the original brass parts. However, up to date, experience with malleable or cast-iron fittings assures satisfactory length of service and should not constitute a problem for the operator. It may be taken as a general rule that wherever brass has had to be eliminated, shorter life can be expected.

■ Vacuum Power Brakes

The general brake picture is good. Brake drums are still being manufactured to the high tensile limits previously specified and the requirements of preventive maintenance apply as previously. Braking parts in general are heavier as cast iron has replaced aluminum and zinc extensively. For example, cast-iron end plates are being used in vacuum power cylinders, which may require slightly more frequent servicing of power cylinders because of rusting which will occur on inside surface. While in many cases tin plating has replaced aluminum alloy in hydraulic cylinder pistons, provided proper plating thickness is maintained, pistons may require only slightly more replacement than the original aluminum parts. Plating in the field is probably not advisable as the flaking which may occur if improperly carried out will cause stickiness in operation. Rubber substitutions have had to be made in parts such as piston-rod guards. Obviously the reclaimed substitute stock will not result in the same life. Furthermore, assembly problems in the field should be recognized, as the tearing of reclaimed rubber when being installed is much more easily done than with original crude rubber. Replacement for vacuum hoses has not been successful. Where such vacuum tubing has been changed by the use of steel, the corrosion problem represents the only serious factor. Lead dipping of the tubing, however, is expected to overcome this difficulty. Synthetic tubing of various types for replacement of vacuum hoses has been tried but these efforts have been unsuccessful up to date mainly because of (1) brittleness of synthetic tubing at low temperatures and consequent danger of breaking if hit by rocks or ice, (2) softening of such tubing at the higher temperatures encountered by the vehicle chassis during warm weather, and (3) difficulty in obtaining a seal between synthetic tubing and fittings except by unusual methods which are not universally available and, therefore, present a serious service and maintenance problem. Synthetic tubing has also been tried in the place of copper tubing, but for the reasons given above, has been equally unsuccessful. Some plastic tubings which have been tried are not even proof against rodents.

With regard to air-brake operation, the situation is, generally speaking, as good as, if not better than the vacuum power set-up because of the different design making necessary the retention of the original materials. A check at the Bendix-Westinghouse Automotive Air Brake Co. reveals that copper tubings are still being used for all connections on commercial vehicles, that their fittings remain in the original brass, and that no change has been made in diaphragms.

■ Lamps

Reflectors are now made of steel as against brass originally used. More frequent examination may be necessary to determine whether the plating on the steel has been uniform.

■ Cabs

Rubber has been eliminated from seats and seat backs, from floor mats and wind laces. Springs of the original characteristics are no longer obtainable for seats and many other parts. These substitutions will result in shorter life and higher maintenance.

It would seem, however, from the overall picture that we are probably not as badly off as other countries in the transportation field, but I think that we should take time to re-examine some of the fundamentals which, to a large degree, can be influenced by the approach of the operators to these problems.

From the engineering point of view, there are several fundamentals that must be emphasized. These may be classified as load and speed; that is, loading per square inch of the wearing parts and rate of movement of those parts. There is a further very important item that we will also have to consider and that is the part which the driver can play in still further prolonging vehicle life after the first two fundamentals have been brought under control.

When we come to the discussion of loads, there probably have been, in the past, economic reasons justifying a certain degree of overloading. The cost of such overloading was not serious as the parts which were affected by such loading could be easily replaced. Today, in a world in which money has ceased to have its usual importance, that is no longer true. Now material is the standard of value that exists everywhere. Under these circumstances the economics have changed so that the situation calls for a reappraisal by the operators of the entire business of trucking in relation to how long the equipment can be usefully employed. At the present time, it could be said that there is no such thing as permissible overloading. The only really safe measure to operate on until this situation stabilizes itself is to take the engineering departments' ratings as the bible, and even some of those may call for a reappraisement in some part of the truck operating units. Obviously this is not always possible.

These facts are not new to operators as they have all been generally known for some time, but it is only now that their appreciation has become vital. A typical illustration would be of the wheel bearings in the axles – this would also apply to any similar type bearing that is elsewhere employed in the vehicle. By and large an axle rating of any given amount allows for a wheel bearing load that would meet the bearing manufacturers' recommendations 100%. In the case of some vehicles, oversize bearings are used that call for an even lighter loading for bearings than the manufacturer would be willing to allow. But if we take 100% as the loading and then proceed to overload the bearing, we begin to see how important the overload can be. If, for instance, an overloading of 20%

on an axle called for a proportionally 20% reduction in life, that might be an amount in normal times not to be regarded too seriously. Unfortunately, that is not the case, and such an overload occasions a reduction in life of such an overloaded bearing of almost 50%. If, with a 100% rating, the bearing has an expected life of 3000 hr, on the 20% overload basis the expected life is only 1600 hr. Conversely, if the bearing is underloaded 20%, the life expectancy is increased a still larger percentage than it was decreased by a similar overload. To give an example that is distorted as far as the original application is concerned, but as an instance of what possibly could happen, if a bearing is only loaded to 50% of its rating, the life is increased from 3000 hr to 30,000 hr – exactly 10 times. It is, therefore, easy to see that between the two extreme examples of 10 times the life on one hand and 50% life on the other, there is room for a good deal of serious consideration when the prolonging of the life of any wearing part is our main consideration.

Tires

Another typical example, of course, is tires, and while it may be already well known, I think it will serve a useful purpose if we again take time to realize that the same story holds true. In connection with tires, we should not fail to take note of the practice that some operators have used to obtain extra tire mileage, namely, installing oversize tires on their axles. If, therefore, the size of their tires is regarded by them as the measure of loading, while they may not be overloading their tires, they can still be overloading the component parts of the axle and, hence, defeating the purpose of conservation. Therefore, the operator who, for reasons best known to himself, has been running larger tires than his axle rating justifies, must revise his scale of loading and return to the actual engineering rating.

The engineer's part is to point out the facts, and it is left to the operator to apply the same economic sense and intelligence that he has used in the past to build up his business in the transportation field, to the business of prolonging the vehicle life in every detail.

I am not losing sight of the fact that this will call for a good deal of hard thinking and a consideration of circumstances which are not entirely under the operator's control. Fleets are a vital part of maintaining war production, and it will be hard to control the loading of vehicles to a point where it will not impede the efficient operation of plants which are serviced by trucks. Nevertheless, we cannot ignore the fundamental importance of the effect of loading and that any overload calls for a disproportionate increase in the life of any part so loaded.

Effect of Speed

Our next consideration is speed. First of all we should analyze just what effect speed has on life of wearing parts and just what gains can be made by a change in operation. It is a fundamental fact that the loads and, therefore, the rate of wear of parts increase as the square of the speed. Under such circumstances, again we have the same disproportion found when considering overloading. A typical example can be used to illustrate this point. If an engine governed at 2600 rpm were to have its speed raised 10% to 2860 rpm, it would have the connecting-rod bearing load increased 21%. We therefore have only to think of all the rotating parts in the power-producing parts of the chassis and envision what change can be made in the operation of the vehicles to effect conservation of each of those parts. We are, therefore, faced with considering reducing the governed speed of the engine, realizing that by so doing we can increase the life of those parts considerably. Again, I realize that such a reduction in engine rpm may affect the schedule of operation to a point where this is not possible but, in any case, until a very careful survey has been made, this factor should be seriously considered. It can also be said that even though it may involve some extra operating cost, the importance of material itself is the main consideration, due to the possible lack of parts for replacement purposes.

Modern truck engines are designed to run at relatively high speeds, and the manufacturers' governed speeds in the past have been based on a standard of wear life which the economics of normal times justified. This recommendation for consideration of a decreased governed speed is again based on economics, but under abnormal conditions. It is important that we see this in its true perspective. Certainly before anything else is done, it is imperative to see that no engine in the vehicles operates above the manufacturer's recommended governed speed. This calls for a very careful checking of governors, and where any operators have felt in the past that for good and sufficient reasons governors could be blocked open or taken off, those governors should be immediately placed back on the engines.

A further pertinent example of the effect of speed is shown by the latest study on tires. Assuming good inflation and proper loading, the reduction in mileage starting with 30 mph at 100% is expressed as follows: 40 mph, 78%; 50 mph, 60%; 60 mph, 45%. These figures are self-explanatory as a guide to the desirability of the lowest possible operating speed consistent with other factors of operation.

Driver's Contribution

All of this leads us into the question of what the driver can contribute. Up to now we have been dealing with inanimate materials.

Our consideration now is of the human material in truck operation. Today additional time, effort, and intelligence must be brought to educate the driver to his part in the conservation of material. It is not unfair to say that a driver may have a perfect safety record but nevertheless his method of operation can be definitely one which may occasion a rapid wearing out of the operating parts of the chassis. It would seem, therefore, that we have to lay down some rules for driver operation as clearly, concisely, and authoritatively as has been done in the past in connection with safety in operation.

The engine undoubtedly is the most important item in the power transmission and is directly under the driver's control from the moment he enters the cab. A tentative set of rules for his strict adherence might be as follows:

1. When starting from cold, the engine should be warmed up slowly. This will allow the oil to reach a good operating temperature and prevent scuffing of pistons. In this connection, he should take extreme care in the use of the choke. Wherever possible, keeping the trucks in a

heated garage will also have its effect in minimizing the injurious effects of starting a cold engine.

2. A very careful check should be kept by him of the engine temperature. When necessary, shutters or covers must be used to keep this temperature at its normal level.

3. The engine speed must be held to a minimum at all times for every particular portion of his run; this minimum, of course, being within good driving range. If the governed speed for the engine is 2600, the engine speed should be held down to around 2000.

4. Reasonable acceleration is also vital. Rapid acceleration definitely should be ruled out. In climbing hills, part throttle should be used, and it is better to run part throttle on the lower gear than to run wide open on the higher gear under loads. No coasting on hills at high speeds, using the engine as a brake, should be permitted. Brake lining will be easier to replace than engine parts. Under no circumstances should the driver run in low gear for any great distance. This gear is designed primarily as a starting gear and not as a continuous-running gear.

Increase Life As Much As 20%

Such precautions as these should increase the life of pistons, cylinders, and bearings possibly as much as 20%, and all this is directly under the control of the driver. It is easy, also, to see where the transmission life and rear-axle life can similarly be prolonged by careful driving. There will be, undoubtedly, many other points that will suggest themselves in over-the-road operation that will still further prolong engine, transmission, and rear-axle life.

Certainly, it can be appreciated that the foregoing suggestions will also have a beneficial effect on tire and brake lining in addition to the gain from careful maintenance of inflation pressure and loads. The amount of rubber that is destroyed by extremely fast starting and severe braking is out of all proportion to normal running wear.

We may, therefore, at this point sum up briefly the main rules, in addition to maintenance, which are under the control of the operator:

1. A careful study must be made to arrange the operations so that only under unusual conditions will the recommended engineering ratings be exceeded.

2. Additional study must be made to rearrange further the schedules so that the engine speed can be held at least 10% lower than at present.

3. The entire thinking of the driver must be focused on his part in this conservation program.

Because the drivers will have to be more than drivers and because they will desire to know how best they can help, there is attached, as an appendix to this paper, a Trouble Shooting Chart. To experienced mechanics and operators it will seem unnecessarily detailed, but to the inexperienced it may prove helpful.

In all this discussion so far, no mention has been made of the part that gasoline and oil play. This omission has been made on the assumption that they are the least important factors, as far as availability is concerned. It is believed that these two commodities will be much more readily available than replacement parts. However, in the overall picture the benefits of the proposed changes in operation must inevitably result in better mileage figures for both oil and gasoline. We should very carefully, in the driver instructions, lay stress on the desirability of avoiding detonation, and on the fact that, where necessary, changes should be made in ignition timing, carburetor, and compression ratio. Excessive detonation, in many engines, may cause cracked cylinder heads in addition to blown out gaskets. What is more important, it may cause serious piston breakage. This breakage, of course, is usually confined to the piston-ring lands.

The octane rating of gasoline for the future is uncertain and changes may come unexpectedly. The driver is the only person who can control detonation, to some degree, by his operation, and report at the earliest possible moment that such control on his part is no longer possible if he is to meet the schedules that have been set up.

Aluminum pistons as compared to cast-iron pistons make possible lower loadings on engine bearings. They are extremely valuable because of the possibility of their having to be replaced in the future with cast-iron pistons. If a change of cast-iron pistons becomes necessary, the engine speeds must automatically be lowered; but even with such lowered speeds, connecting-rod bearings will still be carrying higher loads than desirable. The aluminum piston, therefore, is vital from the conservation standpoint in all operation.

With the available motor equipment getting older every day, it requires more maintenance and not less. Unless the owners of motor transport equipment – through their own service facilities and service facilities made available by the manufacturers of equipment – greatly improve their maintenance methods on their equipment, there will be a serious reduction in the transport equipment available to handle the constantly increasing tonnage requirements that the truck is being asked to transport.

It is a real challenge to both the operator and the manufacturer to build and develop organizations, buildings, and methods that will make certain that every motor truck that is running today will be kept running for the duration and will be maintained at a standard of efficiency that will decrease as much as possible the rate of wear on replacement parts.

Finally, perhaps we should realize that the same strategy and resourcefulness as are necessary on the war front are equally vital on our truck front.

APPENDIX

TROUBLE SHOOTING CHART

This chart is made to provide quick reference for unusual sounds and smells, and for unnatural things the driver of a truck may see or feel as signs of operating trouble. It is divided into four sections: Hearing, Feeling, Seeing, and Smelling. Refer to the section related to your trouble signal. For instance, if a deep regular thump is heard, look under Hearing. The noise and probable cause of such noise is shown.

Under most conditions of truck operation, trouble indications should be viewed as a forewarning of needed adjustment or repairs, and such work should be immediately referred to competent, properly equipped service stations.

Prompt action by drivers will avoid needless expense, costly delays, and will help conserve motor-vehicle equipment.

Hearing

Noise Suggesting Trouble	Cause Discerned by Hearing
Sharp knock when picking up speed.	Distributor not set properly for grade of fuel used.
Light knock when engine is running idle – knock coming regularly at end of each connecting-rod stroke.	Loose wristpin.*
Dull, regular knock in engine in time with crankshaft speed.	Loose connecting-rod bearing.*
Dull, heavy pound in time with crankshaft speed.	Worn or burned out main bearing.*
	Clutch disc not running true.

Light clicking or tapping noise.	Valve tappets out of adjustment.
Intermittent squeal or squeak.	Loose fan or accessory belt.
Continuous squeal or squeak.	Lack of lubrication in generator, water pump, distributor, or other parts.*
Loud exhaust noise.	Muffler or exhaust pipe broken.
Engine backfiring.	Spark-plug porcelain cracked or broken.
	Sticky valve.
	Crossed spark-plug wires.
	Lean mixture.
Popping, spitting, and spark knock.	Valves adjusted too close.
	Exhaust valve head too thin causing hot sections.
	Excessive carbon in engine.
	Weak valve springs.
	Hot spot in cylinder head usually caused by clogged water passages.
	Valves not seated properly.
	Valve timing early.
	Poor fuel.
Running engine misses.	Cracked, short-circuited, or fouled spark plugs.
	Distributor points out of adjustment.
	Rotor contact corroded.
	Battery terminals loose or corroded.
	High-tension cable grounded, or loose connection.
	Low-tension cable grounded, or loose connection
	Loose and short circuited wiring connections to coil and distributor.
Steaming and hissing.	Insufficient water in radiator.
Engine overheats.	Late timing.
	Water pump not working.
	Lack of oil.
	Air cleaner clogged.
	Radiator covered too much.
	Over-supply of antifreeze.
	Plugged or restricted cooling system.
	Loose or missing fan belt.
Engine refuses to stop when switch is turned off.	Excessive carbon in engine.
Engine will speed up but truck will not.	Clutch slipping.
Engine stalls when running idle.	Cold engine. The reaction will be the same as with a lean mixture. Both cause spitting through the carburetor, backfiring through the muffler, and engine stalling.
	Retarded spark.
	Carburetor throttle lever set too low.
	Spark-plug gaps too large.
Steady hum.	Lack of oil in transmission or axles.
	Oil in units too heavy or too light.
	Differential gears worn or not adjusted properly.
Buzzing noise in cab.	Leak in air lines.
Clicking in speedometer.	Dry cable.
Clicking in drive line when engaging clutch.	Insufficient lubrication.
Distinct revolutionary click when turning corner.	Loose wheel.
Dull thud timed with running speed of truck.	Stone wedged between dual rear tires.
	Broken gear tooth in axle.
Body noise. (Rattles, squeaks, sharp snapping, creaks, and sawing sounds.)	Varied causes – loose doors, floorboard, cab support bolts, windshield ventilators, running boards, and so forth.

* Vehicle should not be driven until examined or repaired by a competent service station.

Feeling

Feeling Suggesting Trouble	Cause Discerned by Feeling
Excessive vibration in engine.	Loose engine mounting support.
	Several cylinders not firing properly.
Lack of power.	Loss of compression, valves not sealing.
	Late spark, causing sluggishness and overheating.
	Plugged muffler.
	Incorrect valve timing.
	Hand brake applied.
	Air leaks at intake manifold.
	Slipping clutch.
	Dragging brakes.
	Low or poor compression.
	Fuel pump not functioning properly.
	Air cleaner restricted.
	Cylinders scored or worn excessively
	Weak or broken valve springs.
	Valves sticking in guides.
Hard shifting in old weather.	Oil too heavy in transmission.
Jerky motion in slow speeds or when stopping.	Engine needs tuning, or drive line needs lubrication.
Truck swerves when stopping.	Oil or water on brake linings.
Hard steering.	Tires under-inflated.
	Excessive friction in tie-rod or drag-link joints.
	Abnormal friction in steering-gear assembly.
Low-speed shimmy.	Tires under-inflated
	Too much or too little caster.
	Loose steering knuckle bearings.
	Loose front-wheel bearings.
	Loose or worn steering mechanism
	Eccentric wheels.
	Front spring hangers loose on frame
	Unequal front-wheel camber.
	Weak or sagged front springs.
	Drag link out of line.
	Front brake drums out of round (shimmy only on brake application).
High-speed shimmy.	(Items applicable to low-speed shimmy.)
	Front wheels wobble.
	Front wheels out of balance.
	Ruptured or weak spot in tire, resulting in distortion due to centrifugal force.
	Dragging front-wheel brakes.
Steering wander.	Tires under-inflated.
	Zero or reverse caster.
	Excessive error in toe-in.
	Axles shifted in springs.
	Tight steering assembly or gear.
	Loose wheels.

Seeing

Sight Suggesting Trouble	Cause Discerned by Sight
Sudden drop in oil pressure.	Insufficient oil.*
	Engine overheated, causing thinning of oil.
Excessive oil consumption.	Piston rings, broken, worn, or stuck.
	Piston-ring slots clogged with carbon.

turn to page 279

The EFFECT of INJECTION PUMPS on COLD STARTING

by MAX M. ROENSCH
Chrysler Corp.

"THERE is no satisfactory substitute for the application of heat prior to cranking; without a good heater, reliable starting at temperatures below 10 F is impossible on diesel engines for trucks of the type studied," Mr. Roensch contends.

He explains that this conclusion was indicated in a cold-room investigation of some of the variables affecting the starting characteristics of a Dodge diesel engine, energy-cell – Lanova type, 3.75x5-in. bore and stroke, 331 cu in. displacement. A 24-v starting system is used, he says, which includes a sequence starting switch and solenoids for 6-v pinion engagement and 24-v cranking. An electric air intake heater having a rating of 4350 w at 24 v is also included.

This paper covers the results of a series of cold-room tests to determine the effect of the following factors on cold starting:

1. The quantity of fuel injected at cranking speed.

2. Two types of injection pumps.

Mr. Roensch summarizes his results as follows:

1. For optimum starting, particularly at low cranking speeds, 75 to 85% more fuel is required than that delivered by the pump for maximum power, thus indicating the desirability of having some means of increasing fuel delivery for starting purposes.

2. Fuel pumps must have good distribution, even at low cranking speeds, for unless each cylinder will contribute its best power under these conditions, cranking speeds will not increase as rapidly as they should, and therefore a longer starting time results.

• • •

THE AUTHOR: MAX M. ROENSCH (M '37) received his B.S. degree from Rice Institute in 1926. The following year he attended the University of Michigan. Leaving there with his M.S. degree, he immediately joined the Chrysler Corp. and started under J. B. Macauley, who was in charge of dynamometer work and engine development. Mr. Roensch is now experimental engineer.

OF the many problems to be solved during the development of a diesel engine for use in trucks, cold starting is by no means the least. During the investigation of some of the variables affecting the starting characteristics, it was indicated that there is no satisfactory substitute for the application of heat prior to cranking. Without a good heater, reliable starting at temperatures below 10 F is impossible on an engine of this type. After a satisfactory heater has been developed, tests can be carried on to determine some of the other factors affecting cold starting.

This paper covers the results of a series of cold-room tests to determine the effect on cold starting of the following factors:

A. The quantity of fuel injected at cranking speed.

B. Two types of injection pumps.

These tests were all run in the cold room of the Chrysler Engineering Laboratories on a Dodge diesel engine with the following general specifications:

1. Displacement: 331 cu in.
2. Bore: 3.75 in.
3. Stroke: 5 in.
4. Compression ratio: 14.75:1. Average compression pressure: 300 psi at 100 rpm.
5. Combustion principle: Energy cell – Lanova.
6. A 24-v starting system, which includes a sequence starting switch and solenoids for 6-v pinion engagement and 24-v cranking.
7. Air intake heater: This consists of a coil of No. 9 resistance wire wound in conical shape and mounted at the base of the intake manifold riser such that intake air passes between the coils. The heater is rated 4350 w at 24 v.

For these tests, the cold room was maintained at —10 F, and the engine, batteries, fuel oil, lubricating oil, and all equipment chilled to —10 F for at least 8 hr before starting the engine. The cranking speed was controlled by altering the viscosity of the lubricating oil. For all the tests shown in this paper, the intake air heater was operated for one

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

minute prior to cranking and left on during the starting period.

The fuel oil characteristics were:

a. Distillation: initial, 343 F; 50%, 454 F; and 98%, 544 F.

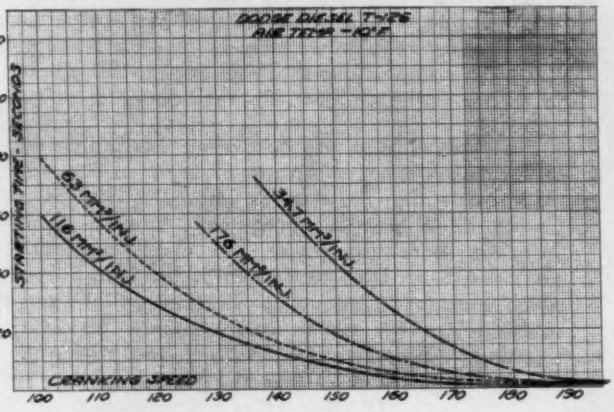
b. Pour point: —20 F.

c. SSU viscosity at 100 F: 32.

d. Flash point: COC, 165 F; TCC, 125 F; fire, 175 F.

e. Cetane value: 50-55.

Using the normal quantity of fuel for full power as a starting point, a series of tests were run to determine the quantity of fuel for best starting. As the normal amount of fuel for full power on this engine is 63 mm^3, the curves in Fig. 1 show the following:



■ Fig. 1 – Effect of fuel delivery on starting

1. Best starting was obtained with 116 mm^3, or 85% in excess of the fuel required for maximum power.

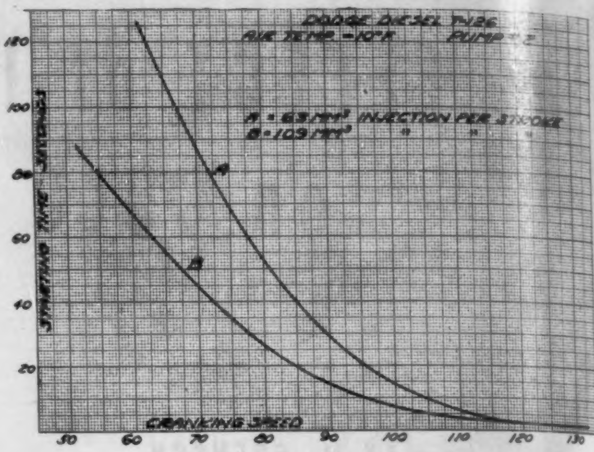
2. Very poor starting was obtained with the lean setting of 34.7 mm^3.

3. It is possible to have too much fuel for good starting, as shown by using the 176 mm^3 per injection, or three times that required for normal engine power.

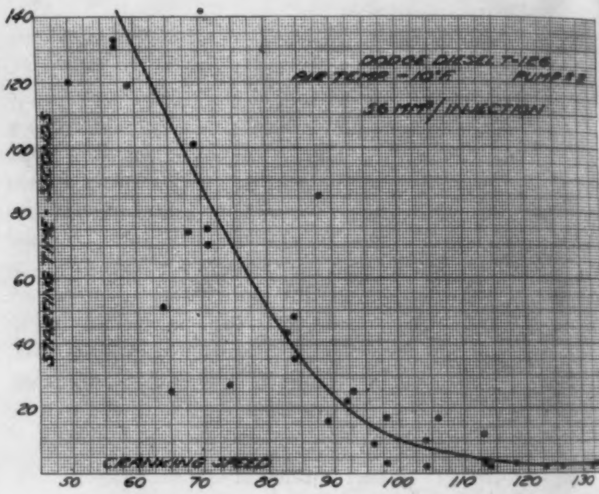
Fig. 2 shows a similar curve on another type of pump, also indicating very clearly the advantage of delivering more fuel for starting purposes. Figs. 3 and 4 show the actual points obtained in making this comparison and are included to show the normal variations that occur on tests of this nature and to emphasize the fact that with increased fuel the starting results are much more consistent, particularly at the lower cranking speeds.

The ability of an injection pump to provide good cold-starting characteristics is an important factor to be considered in selecting the pump for a given engine.

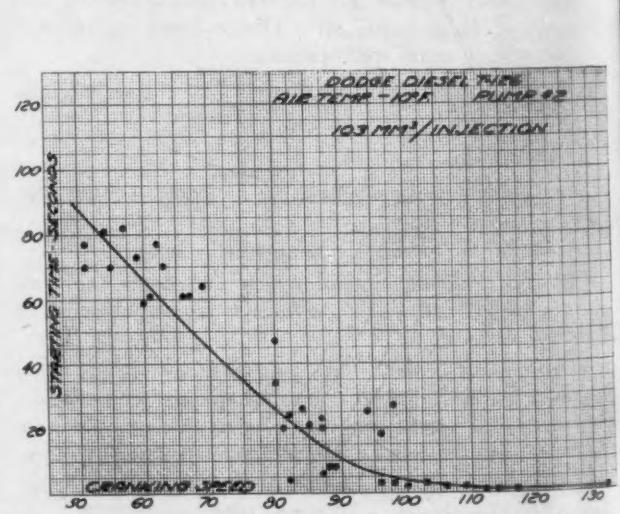
Fig. 5 shows the results of a series of tests on the same engine with two different pumps. Since the two pumps are quite similar hydraulically, it is felt that the better performance of pump No. 2 is largely a matter of better fuel distribution. The maximum variation in distribution of pump No. 2 at cranking speeds is 6.2% and that of pump No. 1 is 25%. These distribution tests were run at room temperature on the pump test stand and observations in the cold room during starting indicate that the difference must be greater under actual cold-starting conditions.



■ Fig. 2 – Effect of fuel delivery on starting

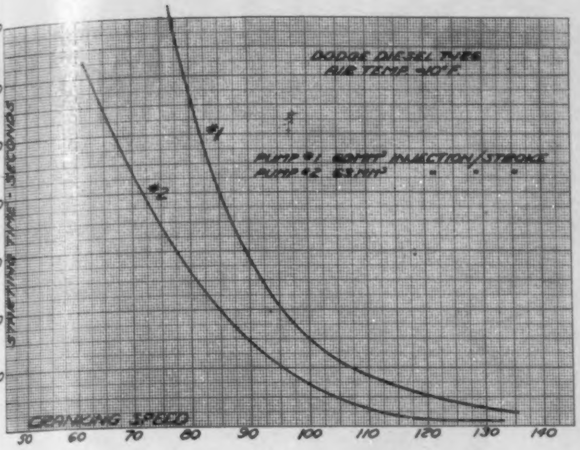


■ Fig. 3 – Effect of fuel delivery on starting,



■ Fig. 4 – Effect of fuel delivery on starting

It is interesting to note that the effect of injection timing on cold starting was not critical. The pump timing required for optimum engine operation was also satisfactory



■ Fig. 5 – Effect of fuel pump on starting

for starting purposes. It is believed this relationship holds true for most engines of this type.

All the above tests were run with the standard production Bosch nozzle. The so-called "throttling" and standard opening type of nozzles, the nozzle opening pressure, various spray pattern values or any special nozzle characteristics had little effect upon starting performance. However, this is considered to be applicable to this engine only, as it is desirable in many cases to have a finely atomized spray pattern for starting. Regardless of the need, the spray pattern values are usually determined by part- and full-load engine operation. Continued research by the oil companies to provide an economically feasible additive for better starting purposes would be considered more desirable.

In conclusion, the results may be summarized as follows:

1. For optimum starting, particularly at low cranking speeds, 75 to 85% more fuel is required than that delivered by the pump for maximum power, thus indicating the desirability of having some means of increasing fuel delivery for starting purposes.

2. Fuel pumps must have good distribution, even at low cranking speeds, for unless each cylinder will contribute its best power under these conditions, cranking speeds will not increase as rapidly as they should, and therefore a longer starting time results.

I wish to express my appreciation to H. H. Dietrich and A. J. E. Roualet of the Chrysler Corp. for their assistance in both the carrying out of the tests and preparation of material for this paper.

Wartime Replacement Parts

continued from page 276

Excessive oil consumption (cont.)	Cylinder bore out of round or excessive taper.
	Cylinder bore scored or badly worn.
	Improper grade and viscosity of oil.
	Oil level too high.
	Oil leaks at gaskets and seals.
Low oil pressure.	Improper grade and viscosity of oil.
	Oil pump screen clogged.
	Excessive bearing clearance.
	Oil pump worn excessively.
No oil pressure.	Not enough oil in engine.*
	Oil too heavy to go through screen in pump.*
	Ice formed on screen in oil pump.*
Black smoke coming from exhaust pipe.	Excessive carbon in engine, or too rich a mixture.
Blue smoke coming from exhaust pipe.	Too much oil in crankcase, or excessive piston-ring wear.
Smoke coming from under hood.	May be short in wiring, or loose fan belt.*
Water or antifreeze lost too quickly.	Overheating.
Lighting equipment fails.	Bulb burned out.
	Short in wiring circuit.
Truck sags at one or more springs.	Broken springs.
Scuffed tires.	Wrong pitch or toe-in.
	Bent or twisted axle.
Spotty tire wear.	Under-inflated tires.
	Dragging brake drums.
	Eccentric or unbalanced tires or wheels

* Vehicle should not be driven until examined or repaired by a competent service station.

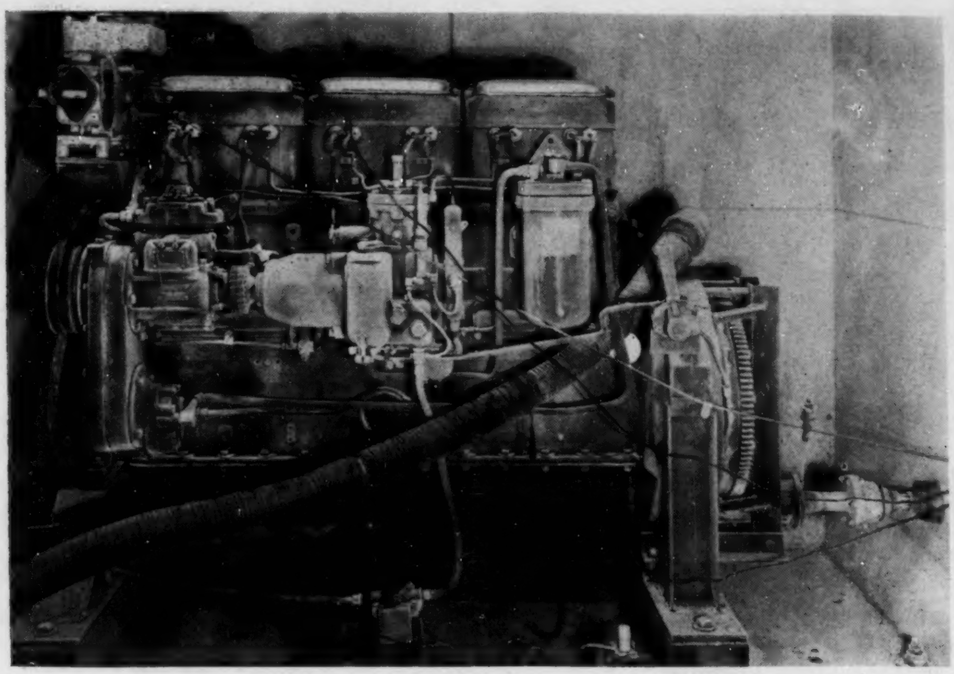
Smelling

Smell Suggesting Trouble	Cause Discerned by Smelling
Odor of gasoline.	Leaky lines or carburetor.*
Odor of burning rubber.	Short circuit in wiring.*
	Wiring fallen on manifold.*
Odor of burning oil.	Clutch slipping.*
	Overheated engine.*
	Dry bearings (for example, on the fan.)*
Odor like burning rags.	Hand brake applied.
	Brake shoes improperly adjusted.

*Vehicle should not be driven until examined or repaired by a competent service station

Difficulties Encountered When the Truck Is Not in Motion

Engine will not start.	Ignition switch not turned on.
	No fuel in gas tank.
	Weak battery.
	Battery terminal corrosion or loose connection.
	Short-circuited or burned spark plug.
	Wet spark plugs or wiring.
	Weak condenser.
	Flooded carburetor.
	Plugged fuel lines.
Starting motor not strong enough to start engine.	Battery weak or completely discharged.
	Loose or corroded connections.
	Engine oil too heavy.
	Engine too stiff.
	Poor carburetor adjustment.
	Excessive choking of the engine.
	Starting-motor commutator burned or dirty.
	Starting-motor armature burned.
	Starting-motor switch defective.



■ Fig. 1 – Cummins 6-cyl, 672 cu in. displacement, supercharged diesel engine set up for the cold-room tests

CRANKIN

by H. L. KNUDSEN

Cummins Engine Co.

STARTING diesel engines at subzero temperatures resolves itself into several distinct problems:

1. The fuel oil must be able to flow freely at starting temperatures.

2. Lubricating oil must be available that permits cranking at about 100 rpm at the desired starting temperature with reasonable starting-power requirements.

3. Starting-power requirements must be such that they may be satisfied by batteries of reasonable size and weight.

4. Engine characteristics must be such that ignition temperatures for the fuel available will be reached within the combustion chamber when cranking at minimum starting rpm for the engine in question.

The purpose of the following analysis was to determine the starting-power requirements at various temperatures with different grades of lubricating oils.

The tests were made on a supercharged 6-cyl engine having a cylinder displacement of 672 cu in.

Fig. 1 shows this engine placed in the cold room, in which it is possible to obtain temperatures down to —40 F. The starting pinion shaft is extended through the wall of the cold room and on the outside it is directly connected, by means of a gear transmission and clutch, to a cradled engine (Fig. 2) capable of developing 100 hp.

The engine speed was controlled by a variable-speed governor. By means of this speed control and the transmission gear ratios, it was possible to crank the engine at any desired speed within the range investigated.

When changing lubricating oil, the engine oil was first heated up to about 100 F and carefully drained. Next the oil pan was filled with the desired oil to the high-level mark (6 gal) and the engine run on its own power, 30 min, to make sure the pistons and cylinders were properly covered with a uniform film of the oil to be tested.

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

THE inescapable conclusion of Mr. Knudsen is that an immersion heater is required for the starting of diesel engines at subzero temperatures. "With this method," he says, "the danger of scoring and scuffing pistons and cylinders during the cranking period due to improper oil films is removed. The engine can be started with the lubricating oil most suitable for it at operating temperature. Cranking power requirements are no greater than in the summertime, so the batteries need not be excessively large for winter starting – even the batteries might be kept up to temperature by an immersion heater, and hence a further reduction in weight and capacity would be possible."

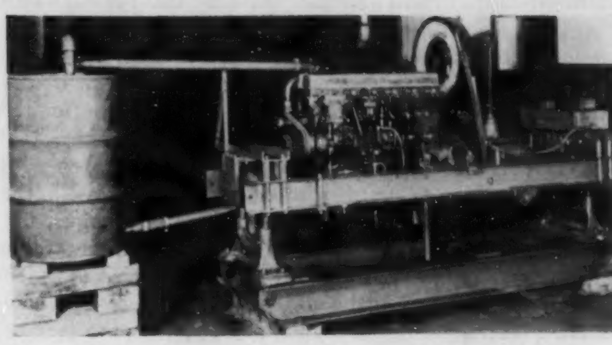
Mr. Knudsen resolves the problem of subzero starting into several parts:

1. The fuel oil must be able to flow freely at starting temperatures.

2. Lubricating oil must be available that permits cranking at about 100 rpm at the desired starting temperature with reasonable starting-power requirements.

3. Starting-power requirements must be such that they may be satisfied by batteries of reasonable size and weight.

POWER and TORQUE REQUIREMENTS at SUBZERO TEMPERATURES



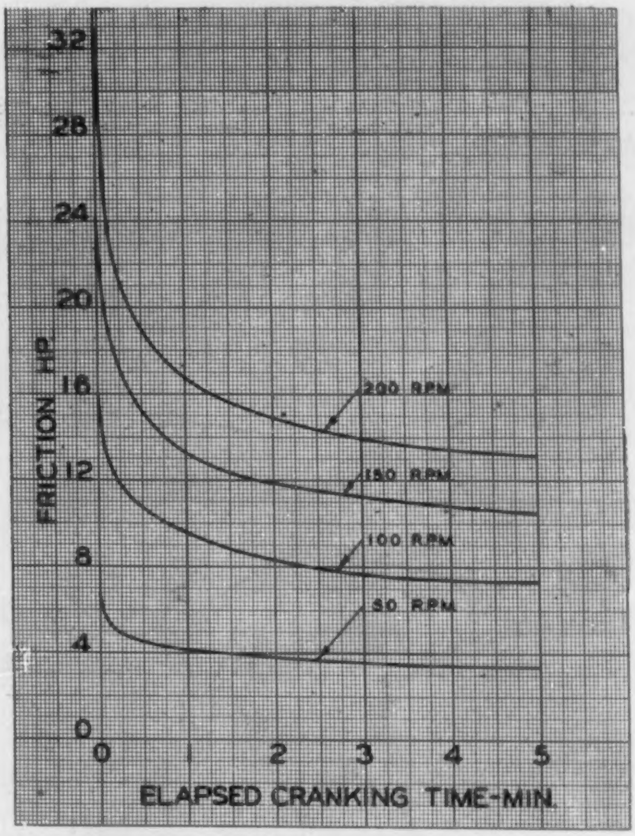
■ Fig. 2 – Cradled engine used to crank the test engine

It was soon learned that the cranking power varied with the cranking time in accordance with a rather definite pattern, the torque always starting at a high value, but falling rapidly at first, until after about 5 min cranking a state of approximate equilibrium seems to be reached (Figs. 3 to 7).

Fig. 3 illustrates clearly the desirability of designing for the lowest possible cranking speed consistent with good and reliable starting.

Examining the curves in Fig. 4A, it is noted that as the temperature decreases the cranking power increases quite in accordance with expectations until we get down to —40 F. At that temperature, while the initial torque is much higher than at —30 F (Fig. 4B), it drops at a more accelerated rate with cranking time and even drops below the —20 F values after about 1 min of cranking. A point of equilibrium is reached after about 2 min of cranking.

The same general picture is presented in Fig. 5 except that here the initial torque is lower at —40 F than at



■ Fig. 3 – Cranking horsepower required at 0 F with oil "A" (SAE 30) – compression released

—30 F. Evidently we are not dealing with strictly fluid friction at —40 F, and consequently the cranking-power requirements are somewhat erratic.

To explore into, and if possible learn, the secret of this behavior, a standard piston with rings was installed in a cylinder sleeve. This assembly was well oiled with "A"

4. Engine characteristics must be such that ignition temperatures for the fuel available will be reached within the combustion chamber when cranking at minimum starting rpm for the engine in question.

Tests were made to determine the starting power requirements at various temperatures with different grades of lubricating oils. These tests indicated that cranking power can be reduced by diluting the lubricating oil with kerosene or similar agents, but such a solution would be unsatisfactory.

★ ★ ★

THE AUTHOR: H. L. KNUDSEN (M '19), SAE past vice-president representing the Diesel-Engine Activity, was engaged in the design and development of the Brons, or – as it is called in this country – the HVID engine, from 1914 to 1922 in the capacity of chief engineer and designer for several HVID licensees. Since 1922, he has been chief engineer, Cummins Engine Co., and in the last few years has devoted considerable time to increasing diesel-engine capacity through supercharging. Mr. Knudsen was born in Denmark, educated in Germany, and came to this country in 1910. He spent a year and a half with General Electric Co., where he was employed as designer of electric d-c motors, before going with the first HVID engine licensee.

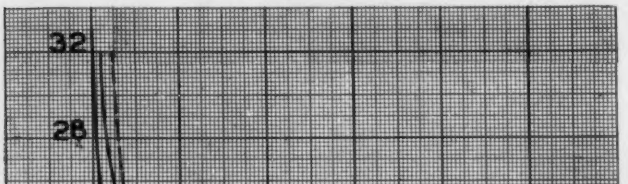
(SAE 10) oil, the same as used in the engine, and heated to between 140 and 150 F. The piston was then pushed back and forth in the sleeve until all surplus oil was removed and only the normal oil film remained. The piston was now removed from the sleeve and a single compression ring installed instead. This sleeve and ring were placed in the cold room and brought down to —40 F.

At this temperature, the ring was gently pushed back and forth in the sleeve with a dummy piston. This immediately created a streaky appearance of the sleeve in a pattern of bright and dull lines, showing that the oil, which at —40 F had a semiplastic and somewhat sticky consistency, was unable to flow and maintain an even film. In other words, we no longer were dealing with strictly fluid friction and this must, we feel, be the reason for the inconsistent behavior recorded at —40 F.

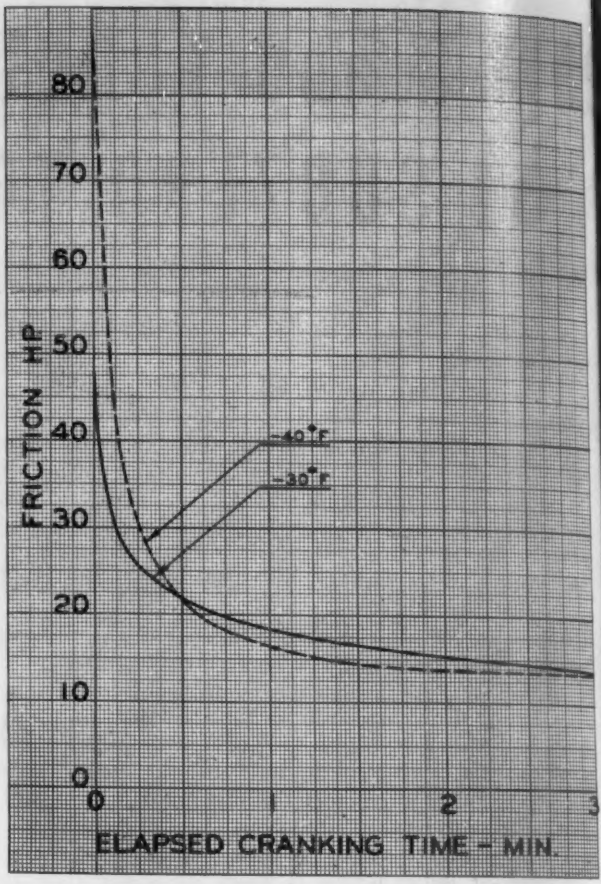
The curves recorded in Figs. 3 to 7 were all taken with the compression released. Instantaneous values with the compression on were recorded also at several points, but since the compression torque has a rather constant value the power has been plotted in a separate curve (Fig. 8) for more convenient analysis. It is interesting to note how small a percentage of the total cranking power the average compression work is at low temperatures.

The "A" lubricating oil used in the tests was a straight mineral oil while the "B" oil is of the additive type, the essential data being:

Oil	SAE Viscosity No.	Gravity, API	S.U. Viscosity at 100 F	Pour Point, F	V.I.
A	30	27.8-28.9	559-580	0	96
A	10	29.6-30.8	195-205	—15	100
B	10	30-32	220-230	—15	103



■ Fig. 4A – Cranking horsepower required to crank at 200 rpm with oil "A" (SAE 10) – compression released



■ Fig. 4B – Cranking horsepower required to crank at 200 rpm with oil "A" (SAE 10) – compression released – continuation of the curves in Fig. 4A

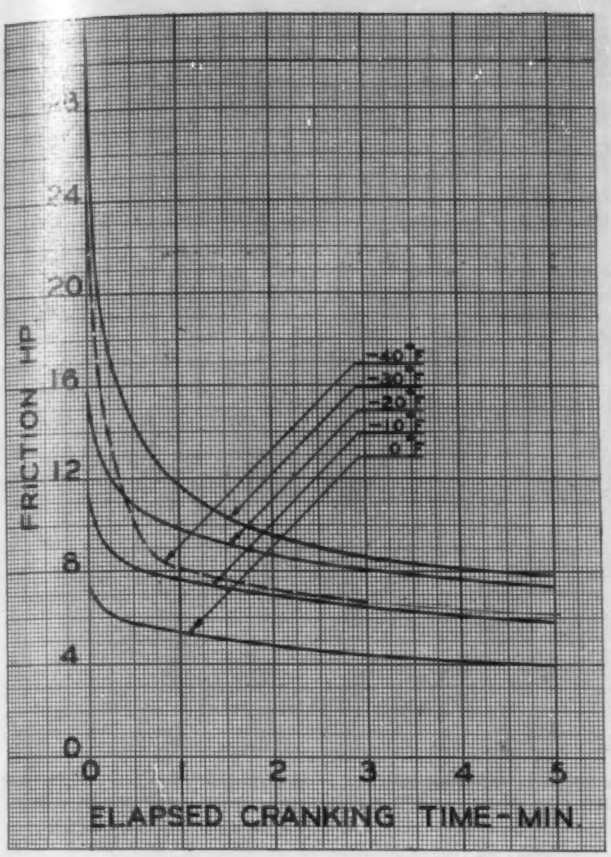
The diluted oil used consisted of 70% "A" (SAE 10) oil + 30% kerosene, and had a consistency at —40 F of about SAE 70 oil at normal temperature (80 F).

A comparison of the data in curves (Figs. 4 and 5) indicates clearly that even with an SAE 10 oil the power requirements for cranking at temperatures below —10 F are excessive and would require starting equipment of impractical dimensions and weight.

To be sure, as shown by the curves in Figs. 6 and 7, the cranking power can be reduced to reasonable values by the simple expedient of diluting the lubricating oil with kerosene or similar agents.

This method, however, is a questionable one, considering the effect of the solution on lubrication at operating temperatures. Using gasoline as a diluent in this way would be a better answer, since the gasoline would evaporate and leave a lubricant of proper viscosity at operating temperature. This would bring on other problems though: Fresh diluent would have to be added for each cold-starting and operating cycle and the diluent would have to be added and mixed while the engine was hot and operated for some time to make sure that the diluted oil reaches and establishes a diluted film on the pistons. Altogether this would be a tedious, expensive, and questionable procedure.

To get some idea of the effect on cranking power of different lubricating oil specifications, a test at —10 F and 200 rpm was run on oil "B" (SAE 10).

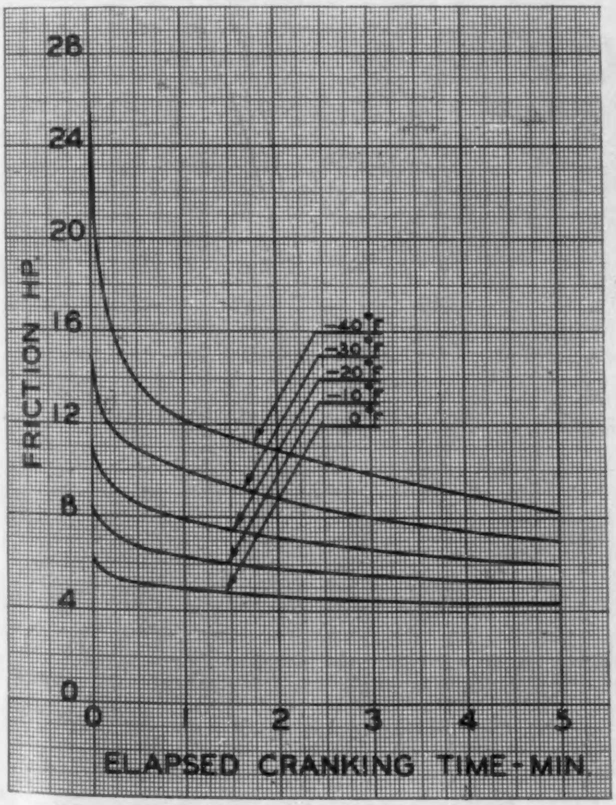


■ Fig. 5 – Cranking horsepower required to crank at 100 rpm with oil "A" (SAE 10) – compression released

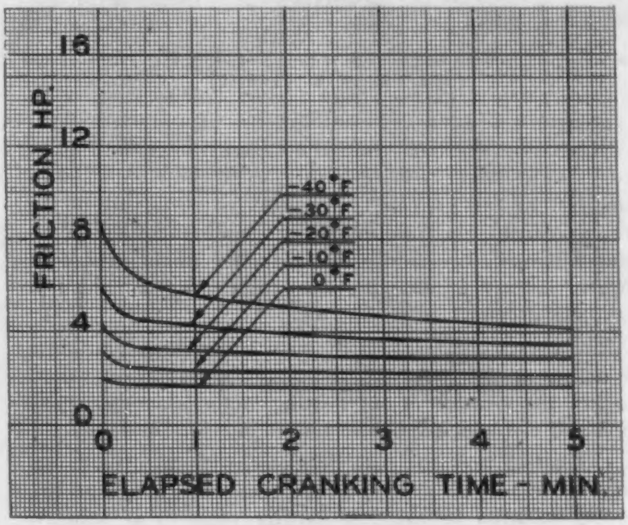
The test was run both with compression on and released to observe also the effect of the compression work in reducing the torque requirements with cranking time. The results are plotted in Fig. 9 and reveal that the compression work has the expected effect, namely, at the start, the compression work is equal to the value (2.2 hp) shown in Fig. 8, whereas after 5 min of cranking the apparent compression work is reduced to 1.2 hp. In other words, cranking with the compression on reduces the cranking-power requirements with time at an accelerated rate.

Comparing now the "compression released" curve in Fig. 9 with the −10 F curve in Fig. 4A, it is noted that the cranking power for oil "B" at the start is about 2 hp higher but it drops at a uniformly greater rate than oil "A" so that after 5 min cranking the "B" oil requires 1½ hp less than the "A" oil.

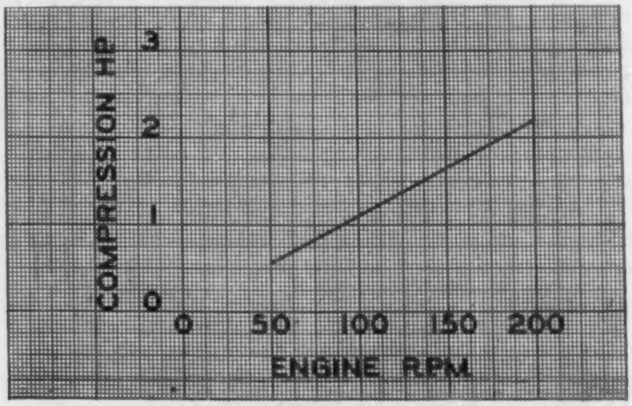
This is interesting, and since the fundamental specifications of the two oils do not differ greatly, the question arises whether the difference in behavior might be charged to the additive in oil "B." Unfortunately, there has not been time to inquire further into this question.



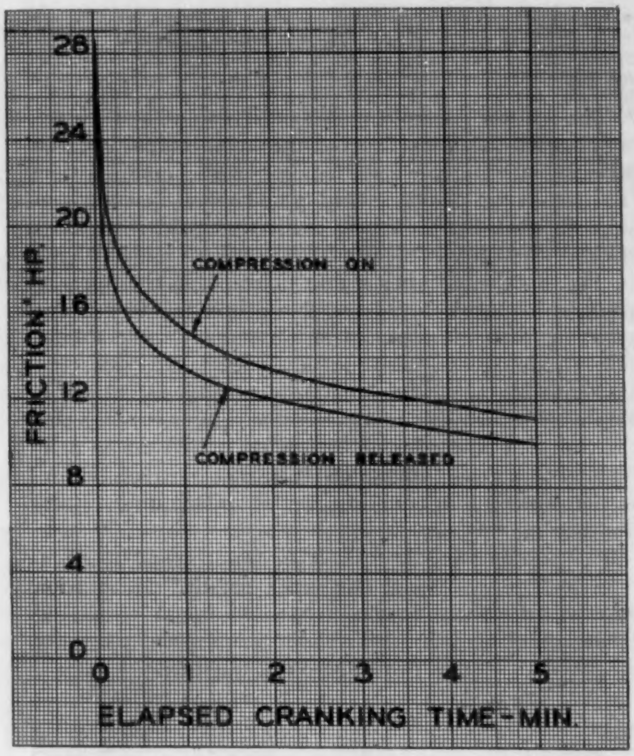
■ Fig. 6 – Cranking horsepower required to crank at 200 rpm with diluted oil – compression released



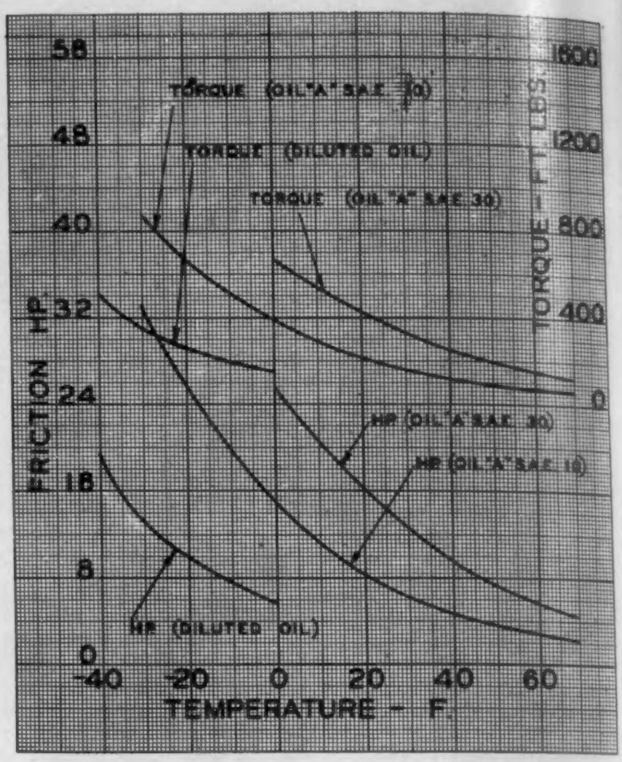
■ Fig. 7 – Cranking horsepower required to crank at 100 rpm with diluted oil – compression released



■ Fig. 8 – Cranking horsepower and torque required to overcome compression – 58 ft-lb constant torque



■ **Fig. 9 – Cranking horsepower required at — 10 F to crank at 200 rpm with oil "B" (SAE 10)**



■ **Fig. 10 – Cranking horsepower and torque required to crank at 200 rpm – compression released – data taken after 5 sec elapsed cranking time**

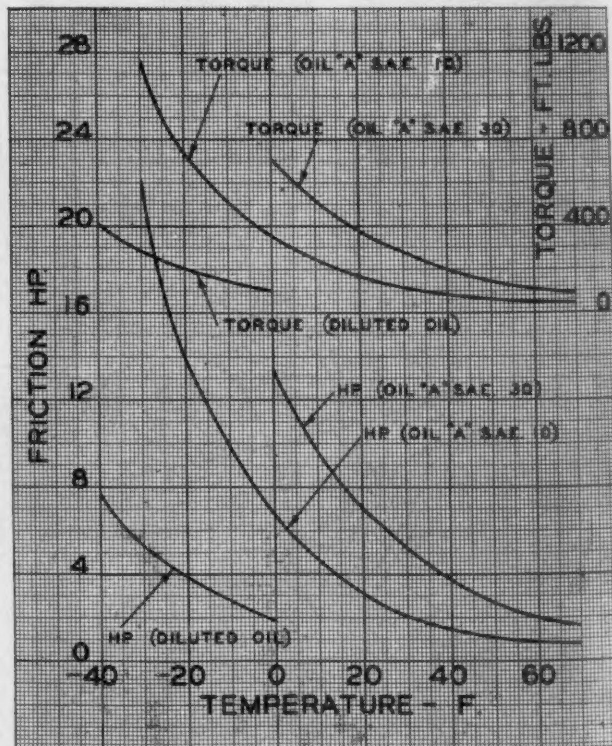
Replotting the data from the curves shown in Figs. 3 to 7, with respect to temperature, as in the curves shown in Figs. 10 and 11, one gets a good visual picture of the rapid increase in cranking-power requirements with reduced temperature; and given a starting motor of known hp, it is easy to establish the temperature at which it becomes necessary to change oil.

The curves are based on the values after 5 sec cranking. Whether or not, however, these are the values on which to base the selection of our starting motor is another question and one which we shall not attempt to answer here.

Evaluating now the data presented with an overall picture in mind, we cannot escape the conclusion that the proper solution to the cold-starting problem at subzero temperatures is an immersion heater.* With this method, the danger of scoring and scuffing pistons and cylinders during the cranking period due to improper oil films is removed. The engine can be started with the lubricating oil most suitable for it at operating temperature. Cranking-power requirements are no greater than in the summer time so that batteries need not be excessively large for winter starting – even the batteries might be kept up to temperature by an immersion heater and hence a further reduction in weight and capacity would be possible.

Lastly, the compression temperature in the cylinders will be higher and make starting sure and snappy. In fact the immersion heater solves all the problems set forth at the beginning of this analysis save one, namely, that of the fuel. Fuels that flow freely at —40 F, however, have been developed so that this problem is now non-existent. So we reiterate that for starting at temperatures below —10 F, the immersion heater is the proper solution.

* Or any other type of heater supplying heat to the jacket coolant.



■ **Fig. 11 – Cranking horsepower and torque required to crank at 100 rpm – compression released – data taken after 5 sec elapsed cranking time**

TERMINAL HANDLING of AIR CARGO

by KARL O. LARSON
Chief Engineer, Northwest Airlines, Inc.

THE handling of the three classes of cargo – unpackaged light pieces, unpackaged heavy units, and packaged goods in different shapes – requires, Mr. Larson says, greater simplification and standardization. Containers used for surface-shipped goods are generally too heavy for air cargo shipping, and lighter paper or fabric covers or containers are desirable. New materials for this purpose, he says, are being tested.

Mr. Larson discusses the advantages of pallet loading with a lift truck. The pallet may be bolted to the floor, thus eliminating strain of lashing.

Parking space requirements for a medium-sized cargo plane are about 150 ft square. A terminal of 10,000 sq ft to handle cargo from this size ship would require a 70-ft wide building one-story high, and 150-ft along one edge of the berth. These figures are based on 12 loadings per day. For each additional berth, the terminal must be extended accordingly. Alternate plan is the large central warehouse for receiving, storing, and transferring. Location opposite the airport is favored, with connecting underground passage. Outlining the interior arrangement, he stresses the need for machine-handling to keep down the time of loading and for the protection of goods.

Due to the variety of plane doors and angles in compartments, the power conveyor system of loading is often a practical solution, where goods are brought to the aircraft. Weather conditions may impede outside cargo handling, so that roof protection is needed. The cargo train is a good competitor of the conveyor system, for it incorporates flexibility of handling for many types of goods, Mr. Larson declares. Fork lift trucks are most adaptable, and with proper ship design, incorporating pallet loading, they offer many advantages.

THE AUTHOR: KARL O. LARSON started his engineering career in the basement workshop of his home in Winthrop, Minn., and in construction camps in Canada, and followed through to graduate with the first aeronautical engineering class of the University of Minnesota in 1930. After a year of post-graduate study he started work with Northwest Airways as a mechanic, but in 1933 went west to lay out the airports necessary for the North Transcontinental Airway. Air transport engineering in all its phases has been his work, hobby and play ever since, and as chief engineer of Northwest Airlines, Inc., he looks forward to much development and expansion in this new field of engineering.

TERMINAL handling of air cargo does not lend itself too well as a subject for technical analysis largely because of the indeterminates that go to make up the principles of the subject. We have no air cargo terminals and we have not yet developed air cargo. We are today handling rail cargo; we are doing it at passenger terminals and hauling it in passenger aircraft.

As of today, we have no air cargo in the true meaning of the word. We carry that small fringe of the total cargo that must move quickly at any cost. We take it or reject it depending on the load capacity of the aircraft. We cannot guarantee its scheduled delivery. Our handling from the time it is picked up until the time it is delivered leaves much to be desired by ourselves and by the shipper.

The whole development of air cargo is now receiving that kind of thinking and planning that will make it a reality when we can cease handling the materials of war and concentrate our efforts and our equipment on transporting the goods of domestic and world-wide commerce. It is because of this formative stage of a new transportation era that I feel it is our duty to sum up our problems and lay down our ideas in such a way that they may either benefit the development directly or do so indirectly by the stimulation of further thought and analysis by others.

Before we can discuss the problems of handling cargo, we must have some idea of what this cargo is to be like:

For the present, we will not concern ourselves with the bulk type of cargo – the unpackaged fluid materials such as the fruits, the grains, the minerals, and the liquids. I grant that the cargo airplane can be built to handle this type of commodity and this type of commodity will find its way into the air cargo picture at an early date, but I hesitate to take time now to discuss the handling of bulk cargo. Until aircraft are built to handle this material in its fluid state, it will be packaged.

There remain three broad classifications of air cargo for present consideration. First, the unpackaged piece cargo that can be stowed, such as lumber, meat, quantity production assemblies, sheet stock, and bar and tube stock, which by its nature does not submit itself easily to packaging or for which a package is of no particular value to either shipper or receiver. Second, the unpackaged heavy cargo for which a crate becomes superfluous, bulky, and heavy. Large machine assemblies, large machine parts, mobile equipment, and furniture are typical of this classification. The third and largest type of cargo is the packaged goods. They may be bundled, crated, baled, packed in drums, in sacks, or in boxes, but however they come or whatever their size or shape, each is a package.

Classifying cargo along other lines, we have the fragile, the perishable cargo, cargo that must be kept dry, cargo that cannot endure freezing, cargo that cannot endure severe cold, cargo that must be kept cool, cargo that cannot

[This paper was presented at the SAE Air Cargo Engineering Meeting, Chicago, Ill., Dec. 8, 1942.]

stand hot sun, the valuable, the inflammable, the explosive, animals, special delivery cargo, and special handle cargo.

Each classification of cargo could require some specialized facility, equipment, or procedure to expedite its handling at the terminal, but it is the object of this paper to review such facilities, equipment, and procedures as can accommodate all classes equally.

The handling of air cargo prior to its arrival at the air terminal may be a function of the air transporter, but at present it is a problem of economics and traffic.

■ New Methods of Packaging Needed

Of considerable concern to us is the matter of the proper packaging by the shipper at his plant. The heavy box, the steel barrel, and the oak keg may be good containers for rail and boat shipments, but their weight and bulk make them expensive and inefficient holders of air cargo. Much thought will have to be given to the design of light-weight yet strong replacements of these commonly accepted containers and, when this has been done, new standards of care in handling will have to be established and practiced to permit their safe use. Treated paper and fabric, thin molded plywood, molded and formed plastics, and the light-weight metals will all find their way into container design. Work has already been done along these lines, and the results are encouraging.

Packaged goods of low density use up valuable space in the hold of any cargo ship to the extent that the shipper must give thought to space-saving methods in his packaging or pay accordingly. Compression baling of fabrics can make efficient bundles compared with loosely packed fabrics in large flimsy boxes and, similarly, tightly baled mail in simple rectangular containers can produce a fire-resistant package easily handled and stowed and occupying small space compared with the bulky mail sack now used. Sponge rubber and many new light-weight inexpensive shock absorbing materials can replace bulky and less effective packing materials used to surround fragile and delicate cargo. Minor disassembly of some products will permit smaller boxes. Stacking of cargo vertically is necessary to utilize the volume of a cargo hold, and doing so adds to the strength requirements of its container.

Until better cargo transport floor designs are in use, the shipper must be advised and educated to limit the unit floor loading of his package so as not to exceed the design strength of the transport floor.

Since this discussion will touch on the pallet system of load handling, it is proper at this time to include a note on container design incorporating the pallet feature. The pallet is a low flat platform raised from the floor several inches by wide runners so spaced as to permit the forks of a lift truck to slide under the pallet and pick up the load and the pallet from underneath. If the floor and runners are made part of the design of a container, the advantages of the pallet are then realized.

Unpackaged heavy cargo can often be mounted on pallets by the shipper, and in so doing he effectively spreads the unit floor loading and simplifies handling. Quantities of small unpackaged piece cargo and small packaged cargo can generally be lashed to a light-weight pallet by the shipper and handled as a large unit of packaged cargo. With proper transport floor design, the pallet can be bolted to the floor, and by so doing eliminates the strain placed on both the cargo and the floor that necessarily exists.

The shipper can further assist the terminal handling of air cargo by stamping the weight of each package on it.

I bring up these points knowing that air cargo cannot be handled efficiently until standards of packing and regulations on preparation of cargo for air shipment have been formulated and have become common knowledge and practice among shippers, just as boat, rail, express, and mail shipments have long been prepared under their individual standards and regulations.

We now have our air cargo prepared and packaged according to air standards on its way to the air cargo terminal, but having no air cargo terminal we had better spend a few minutes surmising the problems and requirements of our airport freight station.

Most of present airports are today undergoing expansion on the drafting boards of the engineers in charge. They are laying in runway extensions, dual runways, taxi strips, and wide-approach clearances, and they quickly realize that the usable periphery of the airport has shrunk to a point where even the extra hangar and the additional space needed at the passenger terminal become hard to provide. What then becomes of the cargo terminal?

We will have to revise our thinking on airport design to accommodate the large amount of space that an air cargo terminal will require, and we will have to give much thought to the cargo terminal so that its space requirement is held to a minimum. This is not the time or place to discuss airports, but a few comments on terminal layout are in order, since the prime purpose of the terminal is to facilitate the handling of cargo. We will disregard the present airport restrictions in discussing terminal design.

■ Terminal Design

Since early phases of cargo operations will necessarily be restricted in volume, it will be important that we start with only a portion of a cargo terminal, so designed that it can expand as the volume of cargo expands without rebuilding and reorganizing each time an expansion is made. Using a medium-sized cargo airplane as the basis for a little quick figuring on terminal layout, we can say that this aircraft can generally be parked in a space about 150 ft square and that it needs one side of the square open to maneuver in and out.

If normally loaded, this ship can hold about 2000 cu ft of cargo, and if this berth were used 12 times per day, 24,000 cu ft of cargo may have to be moved to this position. If cargo did not accumulate for more than one day, the cargo for this berth could be housed in about 10,000 sq ft of terminal or would require a 70-ft wide building, one story high, and 150 ft along one edge of the berth. A two-story building could handle twice the volume, and by docking a cargo ship on opposite sides of the building, twice the volume would be required. This little review does indicate that the terminal area requirements will be sufficient to justify the extension of the terminal for each additional berth added, and that such terminal extension can have sufficient area to handle the cargo volume for that berth.

A long warehouse with aircraft parked on one or both sides will result from planning along these lines. This long warehouse may be one of a group of fingers originating from a connecting header, which could serve either as an enclosed path of communication between fingers or as a central warehouse where freight is cleared to and from surface carriers, where transfer air cargo is handled, and where long-time storage can accumulate. The alternate to

this plan will probably be the large central warehouse for the receiving, shipping, clearance, storage, transfer, and administration of cargo, with delivery to and from the cargo aircraft spotted alongside extended loading docks, underground ramps or lifts, or parked at random in the vicinity of the warehouse. This plan will permit the construction of the warehouse across the road from the already overcrowded airport, with a connecting underground passage. Available airport space, funds for building construction, and operating costs will probably be a greater factor in the early trend of terminal construction than will the small variations in handling efficiencies.

The requirements of the terminal building will be approximately the same no matter how it is built. Rather than discuss these requirements, I will merely list those that are most apparent now, knowing that as time goes on additional requirements will necessarily have to be added:

1. Convenience to airport runway layout.
2. Railroad connection – preferably to transfer yard.
3. Good highway connections.
4. Rail loading docks.
5. Truck loading docks.
6. Receiving room with facilities for weighing and inspection of cargo.
7. Receiving room for mail.
8. Mail handling and rework space and facilities.
9. Dead-storage area.
10. Assembly area for shiploads.
11. Storage area for shiploads.
12. Conveyor system for rapid movement of light cargo and mail.
13. Trucking aisles for rapid movement of cargo trains, lift trucks, and so forth.
14. Pneumatic tube system for rapid dispersal of cargo information.
15. Administration headquarters.
16. Load dispatch headquarters.
17. Airline headquarters.
18. Cold-storage area.
19. Storage vault.
20. Inflammable storage.
21. Communication system.
22. Equipment maintenance facilities.
23. Crating and packaging facilities.
24. Aircraft servicing facilities.
25. Aircraft handling equipment.
26. Aircraft loading equipment.
27. Radio communications to aircraft and control tower.
28. Aircraft air-conditioning equipment.
29. Aircraft ramp and taxi area.
30. Locker room facilities.
31. Aircraft crew lounge.
32. Fire protection.
33. Garages for mobile equipment.
34. Spectator area.
35. Terminal heating plant.

We will leave it to our architects and plant engineers to juggle these requirements to produce the efficient terminal that we must have as an integral part of our air cargo system. They must study the movement of freight to and from and inside the terminal, to and from arriving and departing aircraft; manpower; and communications; and they must evolve a plan that will produce maximum safe speed in the handling of the air cargo aircraft with a minimum of manpower, facilities, and equipment. The actual handling of cargo within the terminal need not vary far from present good practices of material handling where speed, care, and accuracy are given their due consideration. Man handling of air cargo must be displaced by machine handling because of the former's inefficiency, slowness, and the general rough treatment of the cargo itself. Much good handling equipment is available now, and better equipment will come naturally with progress.

In discussing cargo handling from the terminal to the aircraft, we must take a look at the aircraft. This war is probably going to leave us with many good airplanes suited for cargo transport use. Generally, they were originally designed for some other use than the carrying of freight and, though they function as good transports in the air, they leave much to be desired at the cargo terminal dock.

Flexible, Fast, Economical System

Their doors vary in height, width, swing, location, and some are close to the ground and others are not. Numerous small extra compartments exist around the aircraft, each an individual loading problem in itself. Speedy cargo handling inside the aircraft is impossible, and lashings and tie-downs are crude. Around this conglomeration of aircraft, we must build a system of efficient cargo handling. The nature of the aircraft requires that the system be flexible, operations requires that the system be fast, traffic says it must be economical.

Consider, first, the case of the aircraft docked alongside the warehouse where cargo must be moved a distance approximately half the span of the wing, or distances up to about 100 ft. With a substantial building serving as an anchor, it is feasible to lower or swing into place between the cargo door and the warehouse door a powered conveyor belt or track capable of moving the heavy and the light loads either way with man supervision only. At this distance, the variation in door height ceases to be a factor because of the very small positive or negative slope that will result with the terminal end of the conveyor at standard loading-platform height. Using a similar nonpowered roller conveyor, cargo could be moved with a minimum of effort to and from the aircraft by raising or lowering the terminal end of the conveyor to produce slight negative slope in the direction of the movement of the load. Should it be difficult or impractical to swing or lower a powered conveyor into place, as may be the case where the load has been accumulated on a loading dock removed from the warehouse, the conveyor can be rolled into place and elevated to required heights. As long as cargo has to be brought to the aircraft instead of the aircraft going to the cargo, a powered, covered conveyor operating from warehouse to ship has probably more distinct advantages than any other apparent system of handling cargo. Whether or not it will come into use will depend largely on our ability to handle the conveyor.

When the cargo load must be carted to and from the aircraft beyond the protection and facilities of the warehouse, the problems of handling mushroom. The following is a listing of these problems as they apply particularly to the present aircraft and generally to any aircraft.

1. Ramp ice and snow.
2. Multiple handling.
3. Maintenance of control.
4. Maintenance of communications.
5. Protection of cargo from snow and rain.

6. Protection of cargo from freezing.
7. Protection of cargo from hot sun.
8. Protection of valuable cargo.
9. Hoisting and lowering.
10. Illumination for night handling.

Each problem is not particularly difficult to overcome by itself, but too often the common solution of one adds to the difficulty of another.

Cargo handling equipment is generally built to function on smooth and clean floors under the protection of the warehouse roof, and it will function equally well on a smooth and clean ramp on a still, clear day. In some parts of the country, ramps can be kept clean at all times, but in most of this country we do have snow and ice to contend with, and in all places we have wind and rain. In many places, these disturbances are occasional or rare and the accompanying problems are overlooked. *If air cargo is going to function as a scheduled operation, these occasional problems cannot be overlooked in any detail* since the occasional delay on account of snow, the delay on account of ice, the delay on account of rain, wind, heat or cold, when added up, will not be favorable. The general tendency to add manpower to offset the failings of marginal equipment is not economical.

Small-diameter wheels, hard tires, and overloaded pneumatic tires should be avoided. Low axle or frame clearances add to trouble in snow. Carts without brakes roll, and light-weight carts blow over in strong winds. Smooth tires on tractive equipment and light-weight trucks and tractors slip in snow and ice, when their need becomes greatest. Poor wheel bearings and steering mechanisms stiffen in cold weather. All of these factors, and others, are simple design features that can be avoided if the particular piece of equipment is specified to operate with a margin of performance under adverse climatic and ramp conditions. Protection of all ramp load handling equipment from damage, rust, and general deterioration can best be secured by proper design, proper selection of materials, and proper selection and application of finishes. A high standard of equipment design for appearance and maintenance will do much toward a high standard of cargo handling operations.

■ Protection of Cargo

Protection of cargo from the elements during handling between the warehouse and the ship is difficult. Waterproof packaging of all goods subject to damage from moisture would be of greatest help to the cargo handler. The use of cellophane, pliofilm, light rubberized fabrics, treated paper, or waterproofed plywood will go far toward making such a requirement practical. Goods subject to damage by heat or cold must either be packaged with insulation or moved in a conditioned truck to protect such goods for the time they may be on the ramp in any degree of heat or cold that may exist. The use of canvas tarpaulins is generally inadequate and is an interference to fast handling. Enclosure of carts or trucks always restricts the capacity of the unit and generally interferes with handling. Best obvious solution for 100% protection is the elimination of the unprotected run from the warehouse to the cargo door and next best is the minimization of this distance. Overhanging canopy roofs extending out over the loading aircraft are possible and may be practical for the protection they offer.

Multiple handling of cargo is generally necessary when the aircraft is beyond reach of a conveyor directly from the warehouse. An exception to this is probably the mobile elevating platform which can be loaded at the warehouse, moved to the ship either by its own power or by an auxiliary truck unit, elevated to the height of the cargo door and unloaded directly into the ship. If the bed of this platform incorporates rollers for easy movement of the cargo, it then becomes merely an independent movable section of the conveyor system. The cargo train is a good competitor of the conveyor system in almost any problem of load handling. It is handicapped out of doors in bad weather, and it does require multiple handling at the aircraft, but it is flexible, and flexibility of handling operations around the variety of aircraft that are now capable of hauling cargo is a feature of prime importance. The flexibility of this system of gathering and delivering loads presents enough interesting advantages to warrant review.

■ Cargo Train

Units of a cargo train can be dispersed around a warehouse and can be loaded as the cargo accumulates. They make simultaneous loading of mail, express, incoming freight, hold-over freight, and transfer cargo possible.

Loading directly to units of the train eliminates multiple handling in the warehouse. Shippers can deliver their goods already loaded on a train unit.

Units of trains can be assembled at any time ahead of schedule, or they can be picked up at their individual loading positions just prior to delivery to the ramp.

Units holding valuables can be locked in a vault until loading time – those holding perishables can be placed in a refrigerated room.

Trains of any size can be assembled to carry the entire load for a large foreign-bound freighter or a small local pickup helicopter with a minimum of handling equipment and manpower for either.

Transfer cargo can be loaded to and from the cargo train with no intermediate handling, even though any particular carrier's load may be transferred to a large number of other carriers.

Best favored of the cargo train equipment is the low, rubber-tired cart so rigged for steering that it will trail accurately, capable of high-speed operation, easily coupled and uncoupled, and fitted with brakes. At terminals where hundreds of such carts would be required, where a large amount of loaded freight is standing idle, and where shippers will cooperate by loading directly, a simple independent skid platform picked up and moved by a matching hand-lift truck designed with wheels, couplings, and steering similar to that of the cargo train cart is available. This combination equipment effectively cuts down the investment in handling equipment, permits closer stacking in the warehouse, and allows a system of efficient loading in and out of the aircraft.

Many good cargo train towing units are available. Smooth starting, good acceleration, good top speed, good traction, easy steering, small turning radius, and good visibility are essential. The same unit can function in the warehouse or on the ramp with equally good performance.

Hoisting and lowering cargo between the cargo floor and the cargo train platform require powered equipment, except possibly in the case of the low cargo door. A system of hoisting from the top that can be efficient for all classes and sizes of cargo and that can be economically built into

turn to page 304

NEW WINDSHIELD DEVELOPMENTS

by A. L. MORSE

Civil Aeronautics Administration

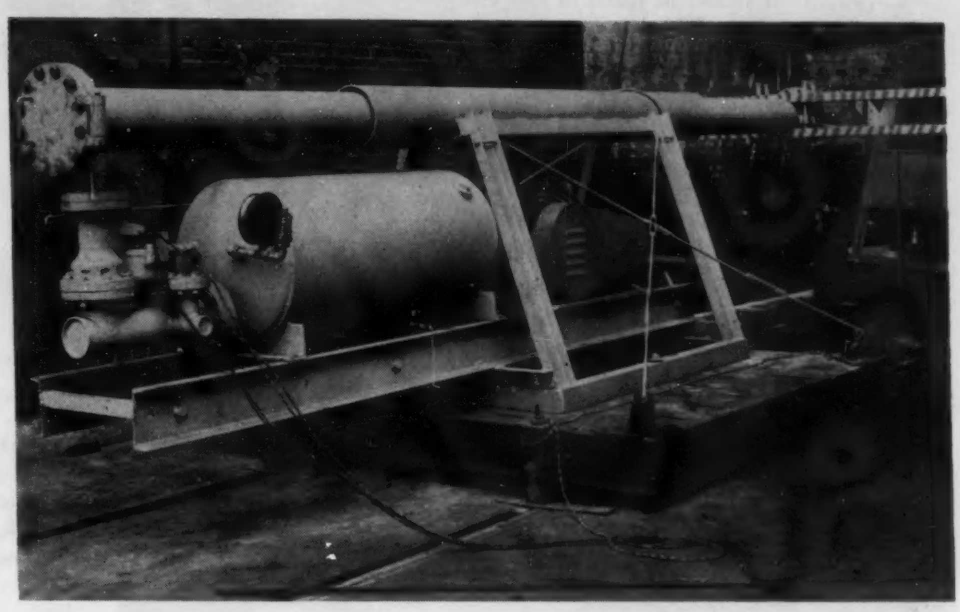


Fig. 1 – The compressed-air gun used in the tests. This gun is capable of tossing a bird carcass weighing 17 lb at a velocity of 400 mph

WHEN the subject of swans versus airplanes was reported upon to the SAE approximately a year ago, the development of means for resisting such impacts existed only on paper. The prospect of obtaining a practical windshield for that purpose now appears much brighter. In the meantime, birds have continued to fly out in front of moving airplanes on the average of twice a week, the conventional windshields have kept on not stopping them, and pilot injuries have mounted. During the past year, some very special bird-windshield collisions have occurred – several hundred of them. In the majority of cases, the windshields were shattered, but no lives were lost. Those collisions were produced artificially to test experimental windshields that were developed to resist such impacts.

The development of birdproof windshields was undertaken by the Technical Development Division of the Civil Aeronautics Administration in July, 1942, and provisions have been made for the continuation of this program through July, 1944. This will include the obtaining of information that generally can be applied to the design of both the windshield and its supporting structure. Furthermore, there is the problem of de-icing, which of necessity must be considered in conjunction with the problem of bird protection since the constructional details involved are interdependent. Several methods for de-icing windshields have been developed by other Government agencies and by the industry. Of these, the circulation of hot air has proved most effective. Therefore the experimental windshields included in the CAA program consist of two panels with a narrow air space between them. Thermal de-icing also is important where bird collisions are concerned, since the maintenance of high temperatures in the plastic interlayers of the windshields has been proved necessary to the maintenance of adequate resistance to impact. Arrangements for cleaning the interpanel surfaces and for installing windshield wipers also must be considered.

The determination of the degree of collision protection which may be deemed adequate is, of course, the concern of the industry and the regulatory bodies within the Civil

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 8, 1943.]

THE development of birdproof windshields was undertaken by the Civil Aeronautics Administration because of the number of pilot injuries resulting when conventional windshields were shattered by birds flying out in front of airplanes – an event that happens about twice a week.

Various panel arrangements were tested to explore the effect of carcass weight, angle of impact, point of impact, panel temperature, method of support, and type of construction.

The test program is still far from being completed, so that no attempt has been made to combine the more promising features in a single windshield that would possess a maximum of impact resistance. The tests thus far conducted have indicated, however, Mr. Morse points out, that the glass vinyl type of laminated windshield with extended plastic edges bolted into an adequately reinforced windshield frame provides practically 100% protection for the DC-3 against collision with birds weighing up to 4 lb, and that a high degree of protection may be provided against collisions with birds up to 20 lb.

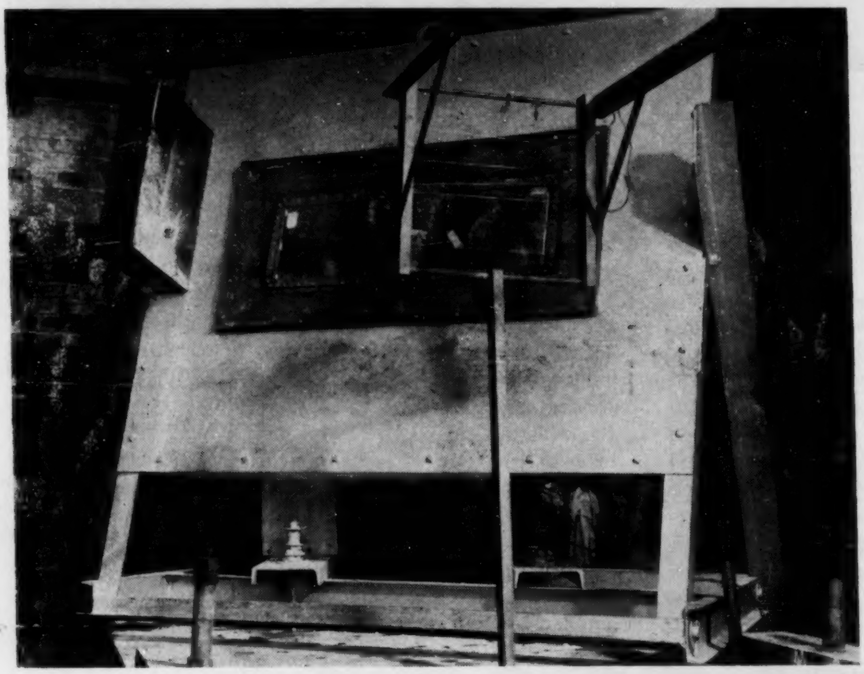
Aeronautics Administration. It is felt, however, that this development program should include determination of the practicability of resisting collisions with the largest birds likely to be encountered in flight and at velocities likely to be realized. It should be noted in this connection that collisions even with small birds will prove extremely serious where the faster airplanes of the future are concerned. Wild swans weigh as much as 20 lb, and airplanes have collided with them.

For test purposes, the projection of a 16-lb carcass is believed to simulate a collision with a 20-lb swan, the wings of which extend beyond the boundaries of the windshield, and for present-day air-carrier type aircraft, impact velocities up to 270 mph are considered likely. Complete facilities for conducting such tests have been provided at the East Pittsburgh plant of the Westinghouse Electric & Mfg. Co. These include a compressed-air catapult or gun, developed by that company, which is capable of tossing bird carcasses weighing as much as 17 lb at velocities up to 400 mph. As shown in Fig. 1, the gun is mounted on a small flat car together with a compressor and an accumulator. Two interchangeable barrels 5 and 8 in. in diameter and 20 ft long are provided. These are connected to the accumulator through an electrically triggered, quick-opening air valve. Pressures up to 350 psi are available.

The experimental windshield panels are located about 9 ft from the muzzle of the gun. These are supported by a larger panel of light sheet steel which buckles under impact, thus simulating to some degree the buckling of the aircraft structures. The sheet-steel panel is bolted into a frame or target of heavy steel construction, which is arranged so that the angle at which the bird strikes the windshield may be adjusted. This arrangement is shown in Fig. 2. The target was provided by the Pittsburgh Plate Glass Co., and is being used to explore the effectiveness of various types of construction, mounting, and so forth. For further testing, the most promising types of windshields will be installed in actual aircraft structures complete with air ducts and with hot-air circulation provided. These installations will be set up in front of the gun in place of the target. Through this procedure, the impact resistance of the airplane as well as the windshield will be ascertained. A preliminary test of the forward portion of a Douglas DC-3 airplane with conventional windshields installed is shown in Fig. 3.

The firing of the gun and the recording of pertinent data are automatic. In a remotely located control room, an electrically operated "gang" of cam-actuated switches open and close the air valve, trip camera mechanisms, and start a nine-channel oscillograph, which obtains strain-gage and accelerometer pickup records, valve timing records, and records the velocity of the bird carcass as it breaks two wires 6 ft apart. A typical oscillogram is shown in Fig. 4. The dynamics of the impacts also are studied through the use of an Edgerton-type high-speed motion picture camera and synchronized illuminators. This equipment obtains 1500 pictures per sec on 35 mm film.

In all tests, freshly killed bird carcasses are used. The birds are painlessly electrocuted, placed in light cloth bags, and are loaded into



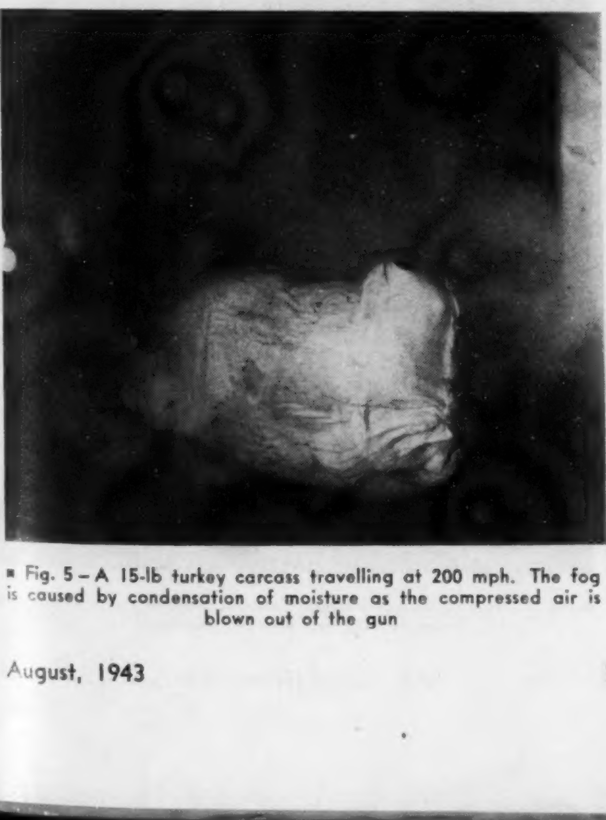
▪ Fig. 2 – (above) Target with an experimental windshield installed. Projectile velocities are measured electrically when the carcass breaks two sets of wires 6 ft apart. The second set of wires is shown in this picture

▪ Fig. 3 – Test of the conventional DC-3 windshield. The carcass is just striking the right-hand inboard panel

▪ Fig. 4 – Typical oscillogram. The strain gage was located on the supporting form adjacent to the panel. The lower trace records the valve operating voltage

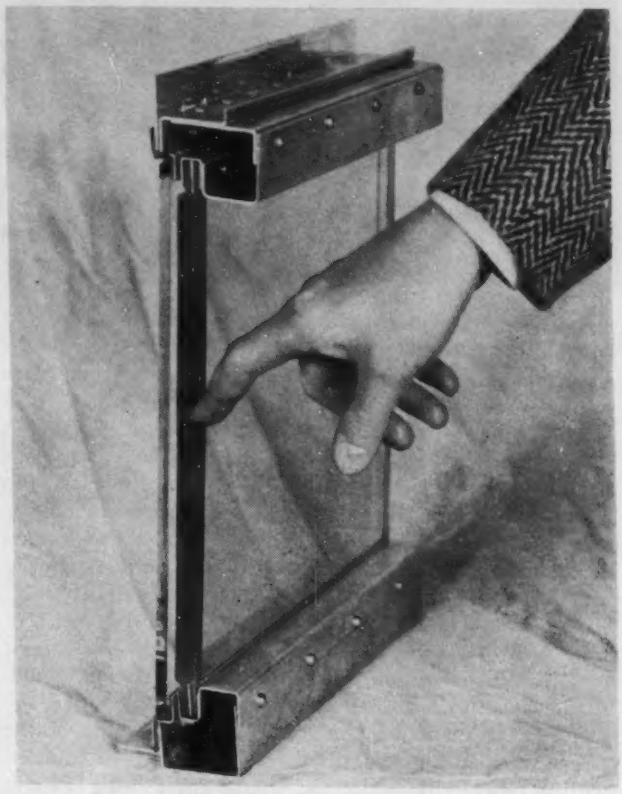
the gun from the breech. The muzzle velocity is governed by the amount of air pressure used and by the distance that the carcass is pushed into the barrel. The use of cloth bags was found necessary to obtain uniform velocities and is believed to have little effect upon the elastic deformations during impact. Chicken carcasses weighing as much as 5 or 6 lb are used to simulate collisions with wild fowl of the same weight. For heavier birds, turkeys are used. The use of actual bird carcasses has proved convenient, and actual conditions are closely approximated. A 15-lb turkey being projected at a velocity of 200 mph is shown in Fig. 5.

The present phase of the program has been influenced by the immediate need for protecting the most widely used present-day air-carrier type airplane, namely, the DC-3. The first consideration was panel dimensions. It was felt, in this connection, that warm-air de-icing had proved sufficiently effective to permit the use of single panels extending from the center post to the corner stanchions, thus eliminating the intermediate posts and sliding panels. Examination of the DC-3 structure provided further confirmation as to the practicability of such an arrangement



▪ Fig. 5 – A 15-lb turkey carcass travelling at 200 mph. The fog is caused by condensation of moisture as the compressed air is blown out of the gun

and also indicated that the height of clear windshield area would not be reduced more than ⅛ in., even when hot-air ducts along the top and bottom edges of the windshield are provided. Fig. 6 shows one possible arrangement which is dimensioned to fit into the DC-3 airplane.



▪ Fig. 6 – Demonstration panel, which illustrates one possible arrangement that incorporates hot-air ducts along the top and bottom edges and is restrained mainly at the ends. The inner panel would be removable in flight. It will be noted that the optical characteristics are reasonably good

Accordingly, it was decided that all experimental panels should be plane rectangles 13⅜ in. high and 37⅞ in. wide, and that these would be employed to explore the effect of carcass weight, angle of impact, point of impact, panel temperature, method of support, and type of construction. For each set of conditions, the basis for evaluation of the impact strength of the panel in question is the minimum projectile velocity that will result in the penetration of the windshield. Usually, the destruction of three or four identical panels will permit the determination of that velocity within ±15%. This is considered adequate for the purpose of this investigation, and the expense and time that would be involved in obtaining and testing a larger number of identical panels for each condition is believed unnecessary. As it is, the total number of panels required is extremely large. Already over a hundred have been reduced to scrap. These have been and are being supplied, without cost, by the Pittsburgh Plate Glass Co., the Libbey-Owens-Ford Glass Co., E. I. duPont de Nemours & Co., the Celanese Corp. of America, and the Rohm & Haas Co. Those companies also have conducted special investigations and tests which have aided considerably in the conduct of the program.

Before testing any experimental panels, it was desired to learn how much protection was provided by the conventional "safety glass" windshield of the size and type

now used in the DC-3 and when supported in the conventional manner by clamping frames. These panels are ¼ in. thick and include a 0.015-in. interlayer of vinyl plastic. Panels of this type were tested both in the target and in the DC-3 structure, and were penetrated by a 4-lb carcass at 75 mph. A typical test of this type of panel is shown in Fig. 3. This, of course, explains why so many collisions with birds have resulted in shattered windshields and injured pilots and further emphasizes the need for the development of stronger windshields.

Panels of similar dimensions but having ⅛-in. semitempered glass faces and a ⅛-in. interlayer of vinyl plastic next were tested. These panels also were mounted in the target and in the DC-3 structure and also were supported in clamping frames. They were pulled out of the frames by 4-lb birds traveling at 90 mph, although they were not penetrated. This test indicated the need for a more secure method of mounting, in order to take advantage of the high degrees of toughness and stretchability that are inherent characteristics of polyvinyl acetal plastic if a proper balance of temperature and plasticizer content is maintained. It therefore was decided that the plastic interlayer should extend beyond the edges of the glass and around the entire periphery to provide means for bolting the panels to the supporting structures. It also was decided to mount de-icing panes ahead of the impact-resisting panels to determine the effectiveness of the combination.

As a result of these conclusions, an extensive series of tests involving approximately 100 test panels was carried out, in which various promising general types of windshield construction were investigated. In this series, the panels were maintained at 75 F, and the angles of the target were adjusted to simulate a head-on bird collision with a DC-3 windshield, the lateral angle being 38 deg from the normal and the vertical angle being 17 deg from the normal.

In all of these tests, except in the few cases noted, a ¼-in. fully tempered de-icing pane was mounted in front of the rear impact-resistant panel with a ¼-in. airspace between. This arrangement is shown in Fig. 7.

The types of rear impact-resistant panel construction included in these tests were as follows:

1. A laminated construction with a vinyl plastic interlayer of different thicknesses, and with ⅛-in. semitempered glass faces. The plastic was extended beyond the edges of the glass on all sides to provide the type of bolted mounting previously described.

With this type of rear panel, numerous secondary features of construction also were investigated, including the use of a laminated glass vinyl de-icing pane, variations in the width of airspace, the use of a metal-strip insert in the extended plastic edge, method of support in the frame, variations in the plasticizer content of the vinyl interlayer, and differences in the glass thickness and arrangement.

2. A ½-in. laminated all-glass construction consisting of two layers of ¼-in. fully tempered glass bonded with a vinyl film of 0.025 in. thickness.

This type of panel was clamped into the mounting frame between rubber strips.

3. Panels of cellulose acetate.

4. Panels of all-plastic laminated construction with vinyl interlayers and face plys of methyl methacrylate.

In connection with types 3 and 4 it was noted that the cellulose acetate and the methyl methacrylate facings did not possess the stretching characteristics provided by the vinyl. These panels were tested in various thicknesses and were bolted into the mounting frame. The cellulose acetate tests, being of an exploratory nature, were carried out without the use of de-icing panes.

Tests involving the panel arrangements previously described are shown in Figs. 8 and 9. They indicated that the laminated glass vinyl construction with extended plastic edges provides by far the greatest impact resistance of the various types of construction investigated. The panels with ⅛-in. vinyl interlayers of 20% plasticizer content were found to resist the impact of a 4-lb carcass at velocities approximating 300 mph. They, however, were penetrated by 15-lb carcasses at velocities as low as 125 mph.

Panels with ⅜-in. vinyl interlayers were found to resist

■ Fig. 7 – Typical panel arrangement. The forward pane is fully tempered glass ¼ in. thick. The collision-resistant pane comprises ⅛-in. semitempered glass faces and an interlayer of ⅜-in. vinyl plastic

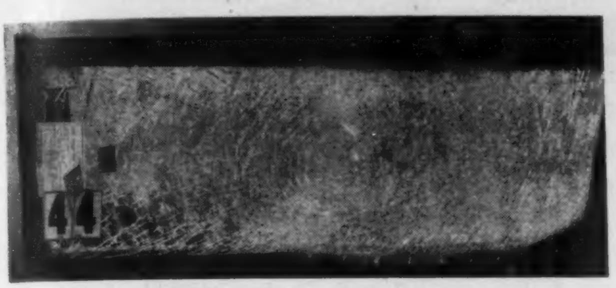
■ Fig. 8 – Test of a conventional safety glass windshield. This shows the result of a collision with a 4-lb bird at an impact velocity of 85 mph

■ Fig. 9 – Test of one of the experimental panel arrangements. The carcass has just hit the windshield

the impact of a 15-lb carcass at a velocity of approximately 200 mph. With a ½-in. vinyl thickness, however, the impact strength was only slightly improved. Panels of all-plastic construction and weighing approximately the same as the ½-in. vinyl glass combination were penetrated by 15-lb carcasses at approximately 130 mph.

In the investigation of the qualitative effect of various details of construction of the glass vinyl type of panels, the following more important conclusions were reached:

1. Vision is destroyed on impact, even though no penetration results. This is illustrated in Fig. 10.



■ Fig. 10 – The rear of a windshield through which there was no penetration. The glass faces craze on impact and destroy vision. This illustrates the need for metal reinforcement to prevent the bolts from tearing out of the plastic

2. A laminated de-icing pane consisting of ⅛-in. semi-tempered glass faces and a 0.060-in. vinyl interlayer, and which is bolted to the rear panel through extended plastic ends, appreciably increases the impact resistance of the combination.

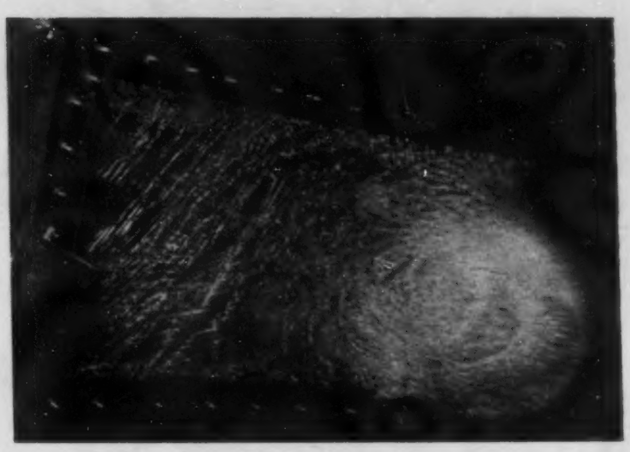
3. An aluminum-alloy strip 0.025 in. thick molded into the extended plastic edge of the panel greatly decreases the tendency for the mounting bolts to tear out of the plastic.

4. Mounting the panel so as to obtain maximum restraint at the ends of the panel and minimum restraint along the top and bottom edges results in increased impact resistance of the windshield but imposes greater forces on the bolts and frame at the ends of the panel.

5. A softer vinyl plastic, utilizing 30% plasticizer as compared to 20% plasticizer, provides greater impact resistance at 75 F.

6. Variations in the width of air space in the panel over a reasonable range had no appreciable effect on the impact resistance of the windshield combination.

One important characteristic of the glass vinyl type of windshield panel revealed in these tests is worthy of special note. It was observed during tests on this type of panel that relatively large quantities of sharp glass splinters are thrown off the rear face of the panel at considerable velocity, even when no penetration of the plastic by the carcass occurs. This is illustrated in Fig. 11. There appears to be no practical means of eliminating this splintering hazard if a rear glass surface on the windshield is utilized. However, tests with the methacrylate plastic laminated on the front and rear faces of the vinyl in place of glass showed that no similar tendency for splintering occurs although, as stated previously, lower impact strengths were obtained. Tests now are being carried out in which a glass vinyl methacrylate lamination is utilized, with the hard plastic on the rear face of the lamination and glass on the front face. The strength and splintering characteristics of such combinations will be determined.



■ Fig. 11 – Pilot's eye view during impact. Although the carcass did not penetrate the windshield, glass is thrown off the rear face. Panels with rear facings of hard plastic are being tested

In addition to the tests previously discussed, investigations also were made of the effect upon impact resistance of the glass vinyl type of panels of several variations in impact conditions which might be realized in actual bird collisions. In particular, determinations were made of the effect of high temperatures upon the impact strength of panels utilizing vinyl plastics of different plasticizer content. Such temperatures would be obtained in a windshield with hot air being circulated through the interspace for de-icing purposes.

It was found from these tests that the penetration velocity of vinyl plastic of 20% plasticizer content is increased approximately 30% as the temperature is raised from 75 F to 115 F. It further was learned that the penetration velocity of vinyl plastic of 30% plasticizer content is decreased as the temperature is varied from 75 F to 115 F.

Although investigations involving the effect of elevated temperatures as yet are incomplete, they are of especial interest, since the physical characteristics of the vinyl plastic are found to vary greatly with temperature changes. The temperature in the plastic interlayer and at different locations in the panel is estimated to vary between 100 F and 200 F when hot air is being circulated for de-icing purposes. This would indicate that a vinyl plastic of 20% plasticizer content or of somewhat lower content should be used to realize maximum impact strength. Furthermore, the circulation of hot air will be necessary whenever ambient temperatures of less than 75 F are encountered in order to maintain adequate collision resistance, since at panel temperatures of less than 75 F, a rapid decrease in impact strength is obtained.

turn to page 304

THE AUTHOR: A. L. MORSE, chief of the Aircraft Development Section of the Civil Aeronautics Administration, began his career in aviation as a Navy balloon pilot in World War I. He returned from the war to finish his engineering training at M.I.T. in 1921. Since that time he has been active in aeronautical work, having been associated during the past 22 years with the following companies: Army Air Service at McCook Field, Glenn L. Martin Co., Fokker, General Airplane Corp., Fairchild Engine & Airplane Corp., Granville Aircraft Corp., Edo, Seversky, and the Federal Aircraft Corp. Mr. Morse joined the CAA in 1937.

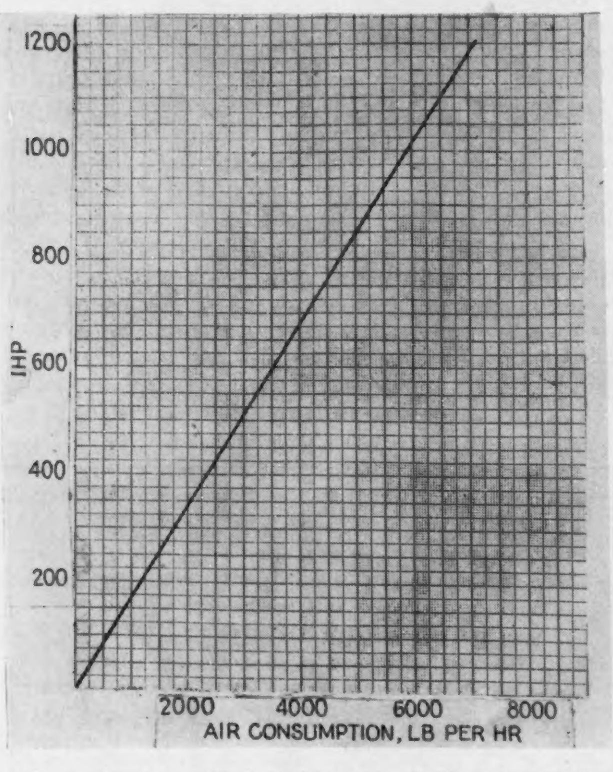
CARBURETION for

by F. J. WIEGAND

Wright Aeronautical Corp.

WITH the advent of high-performance airplanes, able to operate from sea level to substratospheric levels, we have experienced a general improvement in both ground and flight-testing technique of engines and aircraft. Accordingly, a great deal of engineering talent has been concentrated on the problem of obtaining more specific and complete information on the operation of aircraft engines in the airplane over the entire flyable range. This attention to powerplant performance naturally becomes more pronounced as the art progresses, but in the case of high-speed aircraft, it is absolutely essential in order to obtain the most effective airplane. Lately there has been more interest in checking the fuel-air ratios provided for the engine during flight operation, with some attendant confusion as to the results obtained and the suitability of these results. After five years of development of propeller governors, ignition systems, airplane fuel systems, cowling, superchargers and instruments, it would seem that these accessories function satisfactorily enough to enable the carburetor engineer to obtain reasonably good data on the operation of the carburetor in flight. Even now, however, many reported carburetor troubles are traceable to erratic performance of the foregoing equipment; but supposing that these accessories function satisfactorily, what should the carburetor do?

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 12, 1943.]



■ Fig. 1 – Variation of indicated horsepower with air consumption

★ ★ ★

THE importance of obtaining complete information on the operation of engines in the airplane over the entire flying range, particularly in the case of the high-speed airplane, if we are to obtain the most effective airplane possible, is emphasized by Mr. Wiegand.

Mr. Wiegand sets up certain characteristics of aircraft engines that affect carburetion. These are:

1. Since the engine operates on air, the carburetor must meter the fuel in proportion to the weight of air burned, which has a direct relation to indicated horsepower output.

2. Engine air consumption and, therefore, fuel consumption, bear no fixed relation to brake horsepower or brake specific fuel consumption.

3. Since the carburetor meters on the basis of airflow, there is no fixed relation between carburetor metering and brake specific fuel consumption.

The aircraft carburetor has certain functions, which can be summarized as follows: It should meter and mix fuel in a selected proportion to the weight of air consumed by the engine; provide fuel-air ratios for all flight operations; incorporate an acceleration system that will allow the pilot to open and close the throttles at any rate, load, altitude, and frequency; meter satisfactorily with many types of fuel systems and air systems at any airplane attitude.

If such severe conditions are to be met, it is obvious the highest skill is required. Even when the best techniques of today are followed, it cannot be expected that the present carburetors can meter to closer limits than $\pm 5\%$ over the entire engine operating range.

"Considering the variables involved," Mr. Wiegand concludes, "the quality and functional performance of present carburetors are considered satisfactory from a practical operating standpoint, and with further cooperation of the aircraft, carburetor, and engine manufacturers, should continue to improve in a normal manner."

the AIRCRAFT ENGINE

Basic Engine Characteristics

Let us examine, briefly, the characteristics of an aircraft type of internal-combustion engine that affect carburetion. As with all engines, the brake horsepower (bhp) generated depends on the amount of air burned. When relating the air consumed to the power output, it is, therefore, necessary that this air be measured in terms of pounds, not cubic feet. Since the temperature and working stress in the cylinders are determined by the indicated horsepower output of the cylinders, the amount of air consumed can be directly related to the indicated horsepower. Thus, we find that the weight flow of air per indicated horsepower per hour would be approximately 5.9 lb for mixture ratios reasonably close to best power. This value would not change with the mechanical efficiency of the engine, and would not be affected by the type of accessories mounted on the engine. However, if we try to relate the air burned to the brake horsepower output, it has been proved that the weight of air per brake horsepower-hour may vary from 6.6 to 7.4 lb. This variation is caused by differences in mechanical efficiency plus the effect of power-consuming accessories which are mounted on the engine.

From this analysis, we may say that:

1. Since the engine operates on air, the carburetor must meter the fuel in proportion to the weight of air burned, which has a direct relation to indicated horsepower output.
2. Engine air consumption and, therefore, fuel consumption, bear no fixed relation to brake horsepower or brake specific fuel consumption.
3. Since the carburetor meters on the basis of airflow, there is no fixed relation between carburetor metering and brake specific fuel consumption.

If we plot the airflow required per indicated horsepower for any particular engine, we will have a curve similar to Fig. 1.

Thus, for any airflow or indicated horsepower on this curve, we would require a certain amount of fuel in proportion to the amount of horsepower developed by the engine. Inasmuch as this engine is not a theoretical engine, but a practical one, it involves certain compromises, and cannot run at its best power mixture throughout its operating range; likewise, it cannot run at the mixture which provides best thermal efficiency throughout its range. Therefore, the mixture strength required should vary depending on the working condition of the engine; and to illustrate, a typical fuel-air requirement curve is shown in Fig. 2.

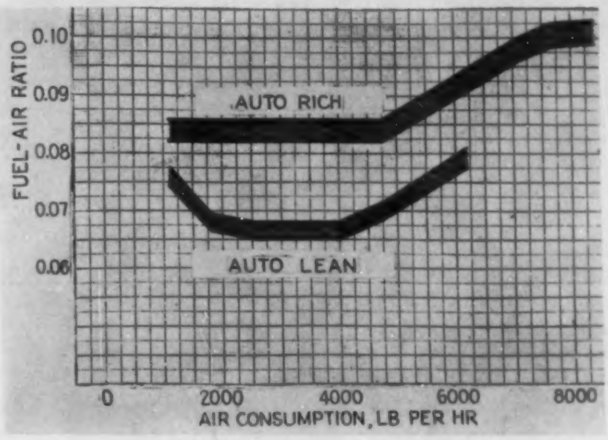


Fig. 2 – Variation of fuel-air ratio with air consumption

Considering this curve, we note two mixture ranges which must be used, and these are termed rich and lean. It is necessary that the rich cruising-power mixture range be provided to allow satisfactory cooling of the engine during flight at critical airplane altitudes, that is, climb, at a high atmospheric temperature. Likewise, the mixture is rich in the high-power range in order to avoid high cylinder temperatures and resulting detonation. The lean range is provided to assure satisfactory fuel consumption when cruising in level flight. You will also note that the rich and lean mixture ranges are restricted by suitable tolerances which, in this case, happen to be $\pm 2\%$. The application of the $\pm 2\%$ limit does not mean that the engine will not run with wider tolerances, but these close tolerances are desired to obtain the utmost in performance, including fuel economy as well as power output over the entire operating range. To pick a specific case, if the mixture at take-off horsepower varies more than $\pm 2\%$ with the usual type of aircraft engine operating at a rich mixture in order to cool satisfactorily during take-off, there will be an appreciable change in the power if the mixture goes too rich and, on the other hand, there will be an appreciable increase in the cylinder temperatures if the mixture goes too lean; likewise, in the cruising range, deviations from the $\pm 2\%$ limit in the lean range can cause appreciable changes in the thermal efficiency of the engine, which, in turn, will affect the airplane range. Deviations from the fuel-air limits in the idle range result in torching or backfiring and poor acceleration, all of which troubles are not considered permissible today. Consequently, through experience with engine operation over the past

THE AUTHOR: F. J. WIEGAND (J '37) now holds the position of senior project engineer in charge of fuel metering projects at Wright Aeronautical Corp. After graduation from Yale University in 1933 he was employed by the National Theater Supply Co. selling movie projects and installing sound equipment in theaters. This did not involve enough engineering, and in February, 1934, he joined Wright's experimental test engineering department as observer and tester. He then became test engineer on carburetion and later field engineer doing experimental flight test work. One of the more interesting flight test projects included the test of the Boeing 314 Clipper airplane project, on which he worked from before the first flight to time of completion of the survey flight to China and return. After about three years of field engineering work he returned to the experimental engineering department and did some research on fuel injection. He became assistant project engineer on carburetor designs, and from there progressed to his present job. Mr. Wiegand is chairman of SAE Committee E-1 on Carburetor Installations.

five years, it has been found necessary to require that carburetors meter to a curve similar to Fig. 2, with the tolerances shown.

■ Development of Ideal Carburetor

Now let us suppose that an engineer, totally unfamiliar with present carburetors, were assigned the job of making a device to meter to these $\pm 2\%$ limits over the range shown on the previous figure without any limitations as to the space or weight of the unit. It is conceivable that he would start by obtaining the most accurate air-metering equipment available. Choosing a hypothetical 1200 bhp engine as the one to which the equipment was to be adapted, he would discover that the airflow varied from 100 to 8000 lb per hr. After diligent search of existing reports and test data, he would shortly realize that there exists no adequate standard to check the accuracy of the air-measuring equipment which he might choose for the job. Consequently, he would have to start by comparing existing equipment for measuring airflow against some arbitrary standard such as a plate-type orifice or a displacement-type meter. Since the airflow range involved is quite large, using the displacement-type meter to check this air-measuring equipment would not be feasible, and he would be forced to check against the flat-plate orifice or some other arbitrary standard.

Recent tests by a Government agency have demonstrated that existing types of air-measuring equipment, used by the aircraft-engine manufacturers and aircraft testing laboratories, when compared to an arbitrary standard consisting of a plate-type orifice, provide reasonably accurate means of measuring airflow. After consulting this data, it is conceivable that our engineer would select one of the various types of air-measuring equipment for his ideal carburetor. If he were to select the type currently used by aircraft-engine and carburetor companies, this equipment would consist of an air bottle equipped with nozzles, varying in diameter from 1 to 7 in. and a suitably calibrated draft gage, as shown in Fig. 3.

The draft gage is connected to the air bottle, and measures the depression in the bottle created by the air flowing through the nozzle. The draft gage is calibrated to read weight flow of air, and is equipped with manual adjustments for barometric pressure and air temperature corrections. The air flowing through such equipment may be calculated by the following formula:

$$W = k \quad CD^2 \quad \sqrt{\frac{Bi}{T_1}}$$

where: W = the rate of airflow, lb per sec

C = coefficient of discharge

D = orifice diameter, in.

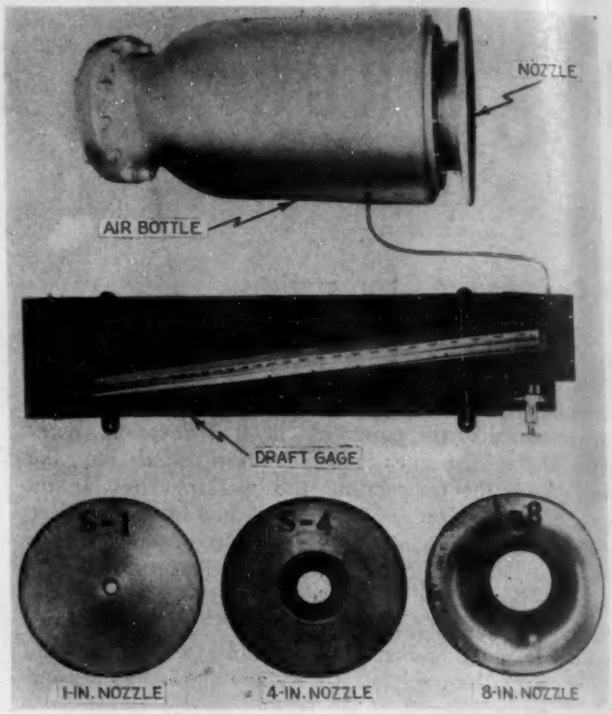
B = barometric pressure, in. of mercury

i = pressure drop across orifice, in. of water

T_1 = air temperature, F absolute

k = a constant, depending on the orifice diameter

It is interesting to note that to obtain good accuracy over an airflow range of 100 to 8000 lb per hr, it is necessary to change the nozzle diameter seven times. If this is done, the test data indicate that these nozzles will measure air weights to an accuracy of $\pm 1\frac{1}{2}\%$ when compared to a plate-type orifice, which is currently used as a standard. This accuracy is obtained under steady-flow conditions, and may not necessarily follow for pulsating-flow conditions.



■ Fig. 3 – Equipment for measuring air consumption

Also, to obtain this accuracy, it is necessary to use a micromanometer, as the standard direct-weight-reading manometer is not sufficiently accurate for this type of work.

Assuming that our engineer has now decided upon the use of the foregoing equipment, since it is as accurate as any other type available, he would now have to select fuel-measuring equipment. There exist, at present, several methods of measuring fuel flow; some of which are more accurate than others. Briefly, this equipment may be listed under the following general types:

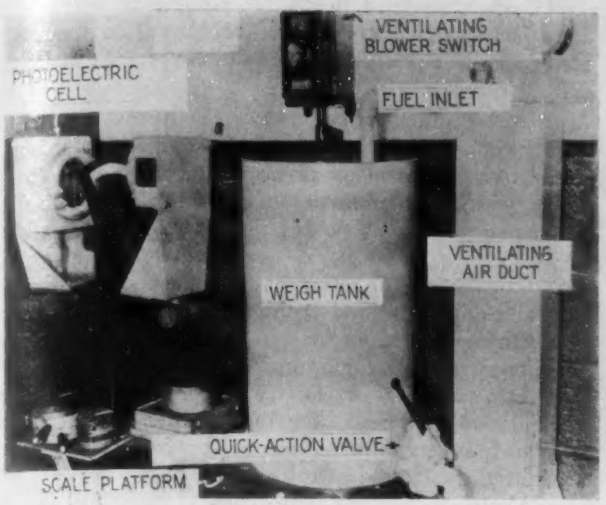
1. Quantity-time meters
   a. Weigh tanks
   b. Volume tanks
   c. Displacement-type flowmeters
2. Direct-reading meters
   a. Rotameters
   b. Reaction-type flowmeters
   c. Displacement-type flowmeters

Since the engineer intends to relate weight flow of fuel with weight flow of air, we can immediately discard the rotameter, volume tank, and both types of flowmeter, since these methods of measuring fuel volume are affected by one or more of the following:

1. Fuel specific gravity
2. Fuel viscosity
3. Fuel temperature
4. Vapor or air in fuel

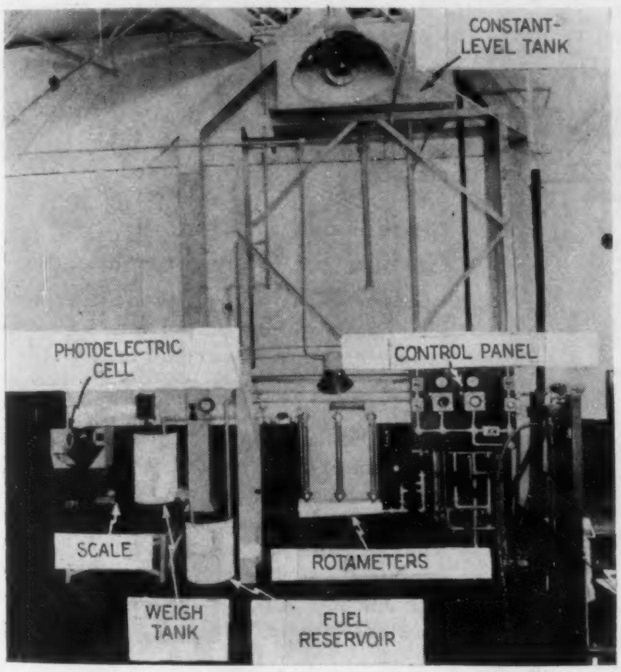
This leaves, therefore, but one method by which we may measure fuel accurately enough to relate the weight of fuel to the weight of air. A typical weigh tank is shown in Fig. 4.

You will note that this equipment consists of a weighing tank placed on a scale with a timer actuated by a photoelectric cell, which, in turn, is actuated by the movement of the scale indicator hand. This type of equipment has been used for some time to calibrate rotameters, and is



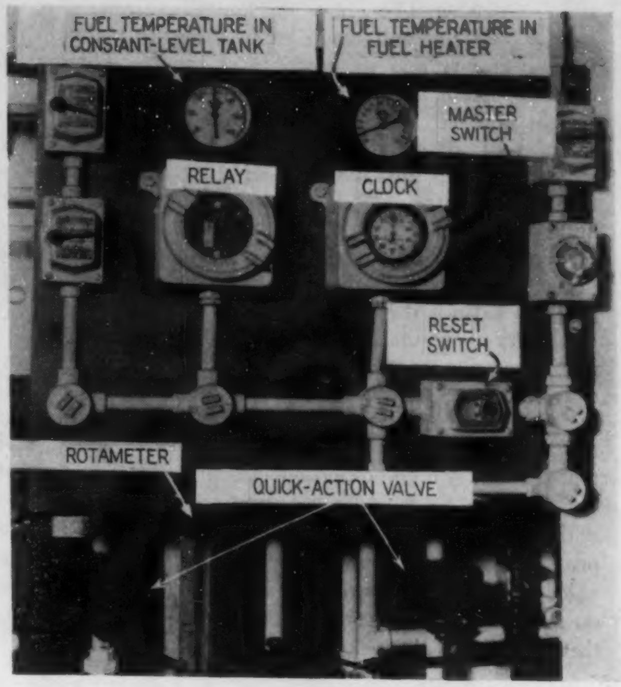
■ Fig. 4 – Close-up of weigh tank and accessories

considered to weigh a volume of fuel within $\pm\frac{1}{2}\%$ regardless of fuel gravity or viscosity. A general picture of the equipment is shown in Fig. 5, and a close-up of the automatic timing mechanism in Fig. 6.

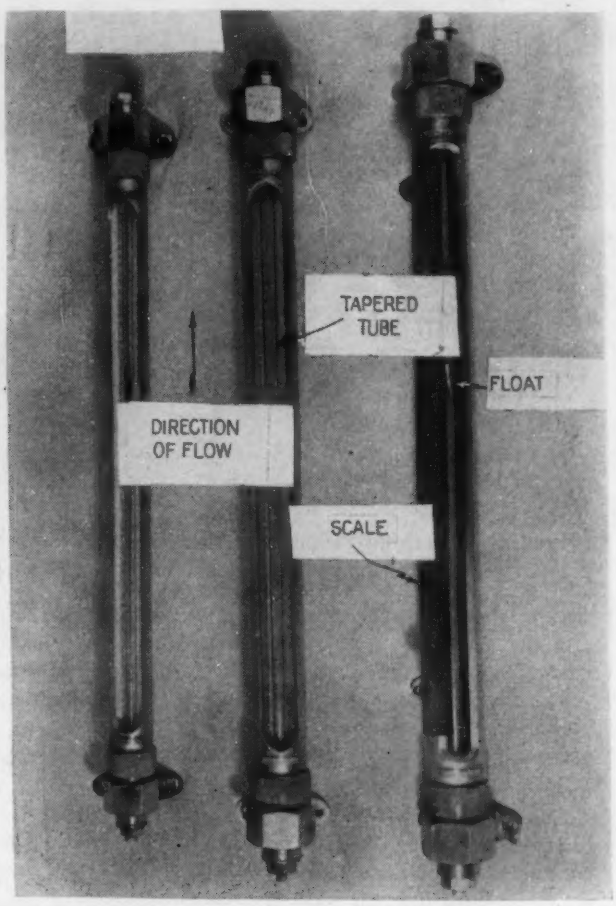


■ Fig. 5 – Equipment for calibrating rotameters

Since this type of equipment is quite bulky, and would be very difficult to use with the air-measuring equipment, because of the complications involved in connecting and coordinating the operation of the weigh tank with the air equipment, it is conceivable that our engineer would compromise and use some type of direct-reading instrument and take the penalty in accuracy. We have had considerable success with rotameters, illustrated in Fig. 7, and since this flow indicator operates on the pressure drop across an orifice, it is possible that this could be used in conjunction with the air-measuring equipment to relate the fuel flow to the airflow. Actually, standard rotameters consist of a tapered tube with a float. The drop across this float is maintained constant for the range of flows for which the



■ Fig. 6 – Automatic timing mechanism

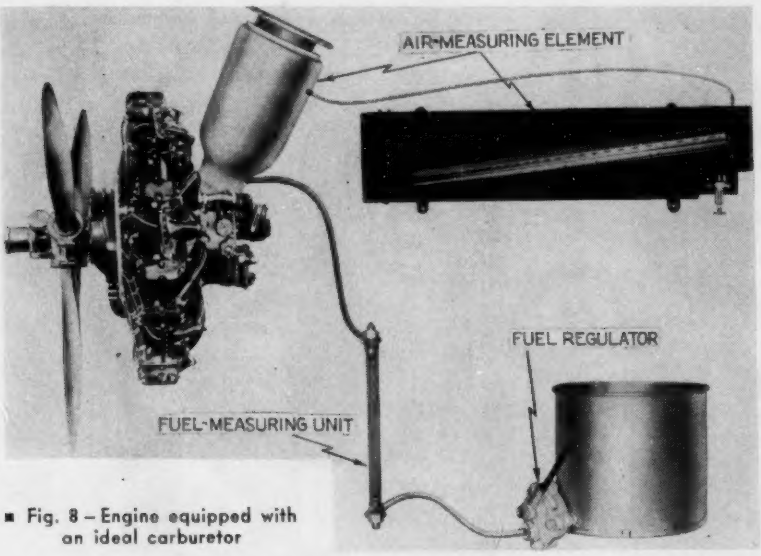


■ Fig. 7 – Rotameters for measuring the flow of fuel

instrument is designed. Thus the float will rise as the volume flow rate of the fuel increases. It is possible that the movement of this float could be connected to an indi-

cator, which would regulate the flow to correspond to the airflow. Of course it would be necessary to maintain the specific gravity and the viscosity of the fuel constant in order to obtain an indication, by this method, of the fuel weight flowing. Also, increasing the pressure drop so as to cover the desired range decreases the sensitivity of the instrument, and thus it would be necessary to use two or three rotameters for a fuel-flow range corresponding to an airflow range from 100 to 8000 lb per hr. It is not quite certain how our engineer would coordinate the indication of the air meter with the indication of the fuel meter, in order to balance the weight flow of fuel and air, but let's not worry about that for the time being.

We now have a method of measuring the air and, possibly, metering the fuel weight in proportion to the air weight. With this equipment, we can safely assume that the air will be measured with an accuracy of $\pm 2\%$, and that the fuel will be proportioned to the air, assuming that our engineer has coordinated the two measuring devices within $\pm 2\%$. These tolerances would then add up to a possible $\pm 4\%$ on the accuracy of the fuel-air ratio provided for the engine airflow. To attain this accuracy requires bulky equipment, and an engine equipped with these devices would look somewhat similar to Fig. 8.



■ Fig. 8 – Engine equipped with an ideal carburetor

Production versions of this metering equipment would probably meter to somewhat closer tolerances. This is caused by two factors:

1. In the transition from the prototype metering equipment to the production units, the checking standard would be changed from the absolute standard (for want of a better word) to a relative standard, consisting of equipment exactly like the metering equipment being checked.

2. Production units merely have to reproduce the mean metering characteristics of the reference unit.

Consequently, we should expect the ideal production carburetor to meter fuel in proportion to the air weight consumed by the engine to limits of $\pm 1\frac{1}{2}\%$ of the fuel-air curve built into the ideal reference carburetor.

This ideal carburetor would have certain deficiencies, such as excess weight, no provision for acceleration or for automatic temperature or altitude compensation, no provision for automatic compensation for variations in fuel specific gravity, fuel temperature, or vapor in the fuel.

Having considered briefly the problem of metering air and fuel as a laboratory procedure, let us now summarize the duties of a practical aircraft carburetor. We desire this device to:

1. Meter and mix fuel in a selected proportion to the weight of air consumed by the engine. Note that this may involve variations in air density from 0.08 to 0.02 lb per cu ft, with any combination of pressures from 34 to 6 in. of mercury, absolute, and temperatures of from 180 to −67 F.

2. Provide fuel-air ratios for all flight operations including take-off, climb, and cruise which will be satisfactory for both cooling the engine and long range with an almost infinite number of engine load conditions.

3. Incorporate an acceleration system which will allow the pilot to open and close the throttles at any rate, at any load, at any altitude, and at any frequency, without backfiring, hesitation, or torching.

4. Meter satisfactorily with many types of fuel systems and air systems at any airplane attitude.

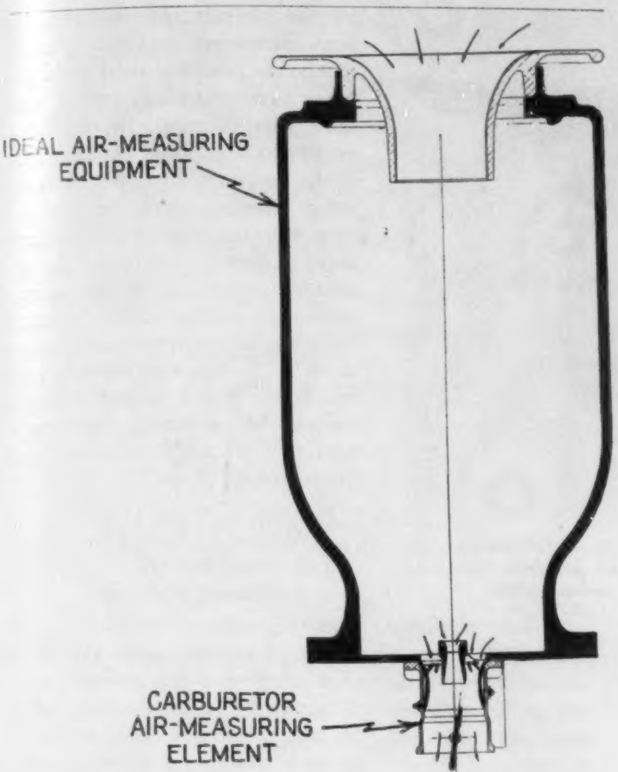
As you might well imagine, the above metering requirements cannot be achieved in practice, and must be qualified and compromised in accordance with existing engineering knowledge. One of the reasons why it is so difficult to meter to close limits is the fact that a practical carburetor should not weigh more than 0.025 lb per bhp or occupy more than 1 cu ft of space. In order better to understand some of the other reasons for the compromises required, let us compare the practical air- and fuel-metering systems to the ideal fuel and air systems previously discussed.

■ Ideal and Practical Carburetors Compared

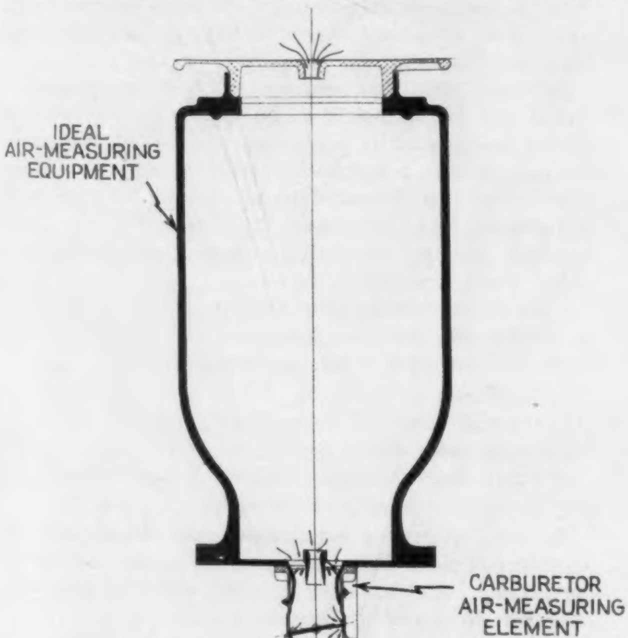
Fig. 9 shows a comparison between the ideal air-measuring orifice and a practical carburetor orifice required for the high-power airflow. Note the great discrepancy between the types of orifices; it is easily apparent from an aerodynamic standpoint that the discharge coefficient of the two orifices would be widely different.

Remembering that with the ideal carburetor the same orifice could not be used to measure the airflow to the accuracy required for both the high and low airflow ranges, let us compare the ideal and practical carburetor orifices for the low airflow range, as shown in Fig. 10.

As you will note, the practical carburetor uses the same size orifice for both the high and low airflow, while the ideal carburetor orifices are proportioned to the airflow in order to obtain the utmost accuracy. Obviously, it is impossible to measure the air consumed by the engine with the practical carburetor to the accuracy obtained by the ideal carburetor, because of the change in the flow coefficient and the high velocities which must be accepted in order to produce a metering unit which can be used on an engine. When one considers that, in addition to the small size of the practical carburetor orifice, there are the additional disturbances created by compromised airscoops, and the necessity for throttles on the lee side of the metering orifice, it seems incredible that present aircraft carburetors measure air as accurately as they do. It is true that a good many of our present carburetor designs show



▪ Fig. 9 – Comparison of ideal and carburetor air-measuring devices – full throttle



▪ Fig. 10 – Comparison of ideal and carburetor air-measuring devices – closed throttle

that not the least attention has been paid to the aerodynamic principles of airflow, but it is felt that this deficiency is gradually being overcome, and should result in a cleaner and smoother airflow path.

A comparison of the fuel-metering systems for the ideal and practical carburetor is shown in Fig. 11. You will note that, whereas the ideal flowmeter incorporates a smooth passage with a good approach to the orifice, the practical carburetor must, of necessity, be far different. Noting on the figure the passage of the fuel through the various jets of the practical carburetor, it is obvious that the approach velocity and flow lines are far from ideal, and it is conceivable that the discharge coefficients of the jets vary considerably as installed in the carburetor. Likewise, some of the jets can affect the approach to other jets, and thus influence the accuracy of metering. It should be noted, however, that in order to attain a compact carburetor, the jet set-up must always be restricted, and again, it is almost unbelievable that this method of metering fuel will compare favorably with the ideal method.

From this very brief analysis and comparison of the basic metering elements of the carburetor, it is apparent that the practical carburetor must overcome severe handicaps to accurate measurement, because of the necessity for condensing the metering elements into a small volume.

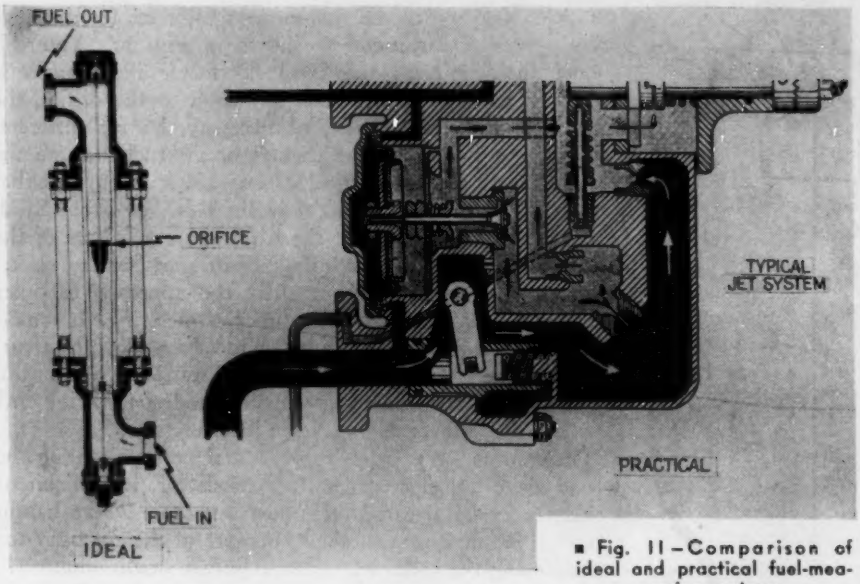
■ Allowable Metering Tolerances

Having described briefly the basic carburetor metering problems, let us now investigate the actual metering performance obtainable with existing carburetor designs. Considering first the reference carburetor, it is important to note that it is precisely set to meter fuel, in proportion to the engine airflow, to provide fuel-air ratios which meet the engine brake specific fuel consumption guarantees in the rich-mixture range. In the lean-mixture range, the carburetor is set to provide fuel-air ratios which approach the maximum-economy fuel consumption as obtained from mixture control curves. Thus, the standards for comparison are the engine guarantees and the brake specific fuel consumption obtained at maximum economy. Existing specifications based on tests over a period of years recognize that it is not possible to set the reference carburetor to the above standards without the application of certain tolerances. These tolerances, allowable at sea level under controlled test conditions, are, for the rich-mixture range, $^{+4\%}_{-0}$ of the fuel-air ratio required for guaranteed specific fuel consumption, and in the lean range, $^{+6\%}_{-0}$ of the fuel-air ratio required to obtain maximum-economy fuel consumptions.

After the metering of the reference carburetor has been satisfactorily determined, additional tolerances are necessary in order to obtain production carburetors. The standard for production carburetors is the reference carburetor. At sea level, existing specifications require metering of the production carburetors within $\pm 2\%$ of the reference carburetor. An additional $\pm 2\%$ tolerance, with respect to the sea-level fuel-air ratio provided by the specific carburetor, is allowed for altitude operation up to 16,000 ft. Above 16,000 ft, this $\pm 2\%$ tolerance is expanded to $\pm 3\%$.

Thus we see that the production carburetors, built to a practical design, will meter to $\pm 2\%$ tolerances at sea level, which compares favorably to the ideal production carburetor which, we have seen, can only meter to $\pm 1\frac{1}{2}\%$ tolerances.

By adding up the tolerances on fuel-air limits, we find that the actual total, compared to the arbitrary (not absolute) standard of guarantees and maximum-economy mix-



■ Fig. 11 – Comparison of ideal and practical fuel-measuring systems

ture strength, may reach $^{+8}_{-4}\%$ in the rich-mixture range and $^{+10}_{-4}\%$ in the lean-mixture range to 16,000 ft. Above 16,000 ft, these tolerances expand to $^{+9}_{-5}\%$ in the rich-mixture range, and $^{+11}_{-5}\%$ in the lean-mixture range.

Naturally, many people, even engineers, would be inclined to view these tolerances as completely unsatisfactory, and out of all reasonable bounds for a metering unit of such importance to the engine and airplane operation. Usually, such critics are not sufficiently acquainted through painful experience with the actual procedure of fuel metering to understand the magnitude of the problems involved and the difficulties attendant to their solution. Also, unfortunately, such critics do not usually have any concrete suggestions to improve the operation of the carburetor which meters to the tolerances previously reviewed, under the following conditions:

1. At all atmospheric air pressures.
2. At all carburetor air temperatures from —20 to 120 F at any carburetor air pressure.
3. With all variations in fuel temperature and viscosities obtained during flight and ground operations.
4. With ambient carburetor air temperatures varying from —40 to 200 F.
5. With any airplane fuel system that maintains fuel pressure at the carburetor.
6. With any airscoop system that can be conveniently mounted on the airplane.

This brief review of allowable metering limits will assist you to understand the reasons why flight-test data obtained with the carburetor are sometimes confusing; and it is, therefore, desired to discuss this phase of the fuel-metering problem next.

■ Flight Test of Practical Carburetors

In the vital struggle to outperform existing enemy aircraft with our own aircraft of stabilized design, the performance of the engine is most important. Since engine performance is considerably affected by the operation of the carburetor, the number of carburetor flight tests run by the aircraft manufacturers has been increasing rapidly. Unfortunately, the results of these flight tests have been confusing, and in some cases disappointing, because of an understandable lack of knowledge of the necessary tolerances on carburetor metering, and the effect of these tolerances on engine performance. Likewise, there has been a general tendency to bypass accepted testing principles, and to overlook the effect of instrumentation inaccuracies on the observed data. In order to effect a better understanding of the variables affecting flight-test results, it is proposed to discuss briefly the following phases:

1. General testing principles.
2. Method of flight test.
3. Instrumentation.
4. Evaluation of results.

1. *General Testing Principles* – Good results during experimental engine tests rest primarily upon the art of measurement. Conclusions relating to the performance of any device stand or fall according to the accuracy of the measurements taken. In this connection, it is interesting to note that there is no such thing as absolute accuracy. For instance, the length of a bar, when measured by a yardstick, might be 4 in., but if a micrometer were used, it might be 4.051 in.; likewise, if an instrument of nicer capabilities were used, the dimension might be 4.0556 in. Thus, to attain absolute accuracy, we must have an instrument with gradations corresponding to infinitely small fractions of a unit.

Assuming that this instrument might be obtained, it would still be necessary to prove that it was accurate within the limit of its gradations, and this would require comparison with a standard of absolute accuracy. Again, this implies that the standard of absolute accuracy must be measured by an instrument of absolute accuracy. Thus, to create such an instrument, it is necessary to have the article which is wanted.

If we do not consider gross mistakes caused by ignorance or carelessness, there are five types of errors that might affect the accuracy of a measurement as follows:

a. Accidental errors.
b. Personal errors.
c. Instrumental errors.
d. Errors due to improper location of instruments.
e. Errors in setting up test procedure.

To avoid the errors noted above, and to eliminate the possibility of these factors affecting the results, the following general rules should be observed when making measurements on variable quantities:

1. All instruments and test apparatus should be calibrated before the test, and calibrated or rechecked afterwards. Instruments which may not maintain their calibrations throughout the tests should be calibrated in intervals during the test.

2. The adequacy of the instruments and apparatus should be determined by preliminary checks before starting the final test.

3. The person in charge of the test should have a sufficient number of assistants so that he will be free to give

attention to the general progress of the test, and to see that the chosen operating conditions are maintained.

4. Equilibrium of all instruments should be obtained before readings are taken.

5. Observations of fluctuating or variable quantities should be made at equal time intervals to secure a true average, and more than one set of readings at the test condition should be taken.

6. It is necessary to make rough calculations on the data as they are accumulated during the progress of the test, to avoid mistakes which would nullify the whole test.

2. *Method of Flight Test* – In order to obtain sufficient data for a good analysis of the operation of the carburetor in the airplane, it is necessary that the following curves and tests be run:

A. Curves and Tests.

1. Idling characteristics.
2. Acceleration characteristics.
3. Flight propeller load curve (ground running).
4. Rated power climb curve to critical altitude in low blower.
5. Rated power climb curve to critical altitude in high blower.
6. Cruising climb curve from 1000-ft altitude to critical altitude.

Note: In the event that sufficient readings are not obtainable in the climb curves of items 4, 5 and 6, supplementary data should be obtained in level flight runs at military rated and maximum cruising powers at various altitudes, including critical altitudes.

7. Full-throttle climb curve in low blower from critical altitude to service ceiling.
8. Full-throttle climb curve in high blower from critical altitude to service ceiling.
9. Maximum cruise readings in automatic lean (AL) or cruising lean (CL) mixture control position in level flight. (Low blower and high blower.)
10. Idling characteristics at 5000-ft altitude (multi-engine ship).
11. Engine operation in glide.

General Note: The climb and cruise curves should be conducted with the heater door in the full cold, full hot, and intermediate positions.

B. Data to be Taken at Each Point.

1. Altitude.
2. Indicated air speed.
3. Rpm.
4. Manifold absolute pressure.
5. Bhp from torquemeter or from manifold-pressure curves if torquemeter is not available.
6. Fuel flow.
7. Specific fuel consumption.
8. Fuel-air ratio from indicator.
9. Fuel pressure.
10. Metering suction (differential gage).
11. Throttle position.
12. Mixture control position.
13. Carburetor air temperature.
14. Cylinder-head temperatures.
15. Fuel temperature.
16. Outside air temperature (take at 100 mph indicated air speed preceding level flight runs).
17. Carburetor ram.

Note: A statement of engine operation as visually observed by the pilot should accompany each point.

3. *Instrumentation* – Before the torquemeter was available, carburetor metering was checked in flight by reference to the exhaust gas analyzer. During many tests, it was found that this instrument was quite reliable in the range of fuel-air mixtures from 0.07 to 0.11, provided the installation of the analyzer equipment was made in accordance with the manufacturer's instructions. Minor errors of temperature compensation in the indicator and the analyzer cell affected the results obtained, but by recording the operating temperatures of the equipment it was possible to correct the observed results and obtain reasonable accuracy. It is considered that this method of checking carburetor operation is still satisfactory, provided the exhaust collector has one outlet and the engine is in good operating condition. With the recent changes in the composition of aviation fuels, however, another correction, based on the hydrogen-carbon ratio of the fuel, is necessary, because it has been found that fuels of high carbon content cause a richer-than-normal indication on the fuel-air analyzer.

With engines equipped with exhaust collector rings having more than one outlet, or using individual exhaust stacks, it is felt that this method of checking the fuel-air ratio is unsatisfactory. Consequently, on most airplanes, it is a normal practice to incorporate a fuel flowmeter to check carburetor metering.

Commercially available flowmeters suitable for flight use may be of either the reaction or the displacement type. Both types are affected by fuel temperature, per cent of vapor by volume in the fuel, and to a minor extent by fuel viscosity. In addition, the accuracy of these meters is sometimes considerably affected by fluctuations in the fuel pumping system or by the action of the carburetor vapor separator. Comprehensive tests of the available meters indicate that the displacement type is accurate within ±2% if properly calibrated, but this degree of accuracy is not always attained because of mechanical difficulties. The reaction-type meters will measure fuel with accuracies varying from ±1½% to ±4%. These tolerances are based on tests made under controlled conditions, and do not apply necessarily to meters installed in aircraft. Changes in fuel temperature from 0 to 120 F introduce errors in the accuracy of the measurement of from 4 to 10%, depending on the type of meter used. Introduction of approximately 10% of vapor by volume into the fuel will cause errors of from 5 to 10% in the indication. Variations in the temperature of the meter elements, separate from the effect of fuel temperature, have not been investigated, but could introduce additional errors. Thus it may be stated that, from past experience, with bench and flight tests of all flowmeters, to expect indications closer than ±3% would be wishful thinking. It is suggested, to improve the accuracy of the results with these meters, that:

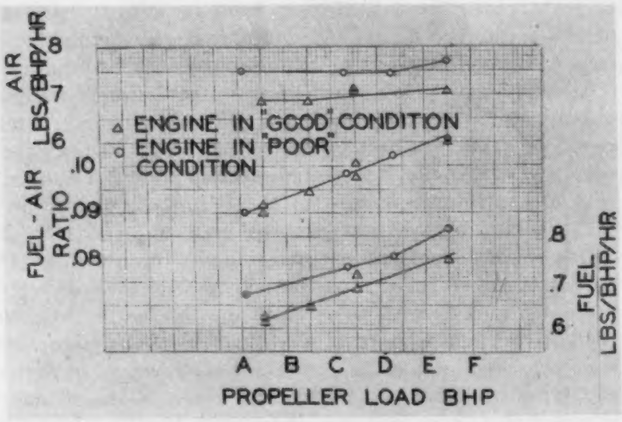
1. A good vapor eliminator be installed in the fuel line before the fuel flowmeter.
2. All fuel flowmeters be calibrated to measure gallons per hour (instead of pounds per hour) and the readings converted to pounds per hour by using the specific gravity of the fuel reaching the flowmeter at the time of the reading.
3. All fuel flowmeters be calibrated in the airplane with the airplane fuel system operating.

The other major instrument used to check carburetor metering is the torquemeter, which, in combination with the tachometer, can be used to obtain an accurate indica-

tion of brake horsepower output. This precision instrument has been of inestimable value in determining how well the engine is performing its duty and how well the aerodynamicist has predicted the performance of the airplane. Data taken over a period of years have demonstrated that the torquemeter will indicate the torque within $\pm 2\%$, provided the indicating equipment is installed properly. *It should be noted, however, that the torque indicator alone does not measure either brake horsepower or indicated horsepower.* Consequently, there is no fixed relation between what the torquemeter indicates and the actual engine output.

In addition to the above major instruments, several pressure gages, thermocouples, and position indicators are necessary, but it is felt that the accuracy of these instruments is mostly a function of the degree of attention to proper calibration and installation, which are indeterminate quantities and cannot be evaluated.

4. *Evaluation of Flight-Test Results* – Since there is no way known to the writer of directly checking the carburetor for metering characteristics in the majority of current airplanes, it has become necessary to use brake specific fuel consumption as a criterion of carburetor metering. *In the first part of the paper, it was proved that this is not a satisfactory means of analyzing carburetor flight data, and it is desired to emphasize this point again.* Fig. 11A shows the comparison between brake specific fuel consumptions obtained with an engine in good and poor condition for the same fuel-air ratio. Note the wide discrepancy in brake specific fuel consumption. In addition to the above comment, the following specific factors must be thoroughly understood and considered, when analyzing flight data:



■ **Fig. 11A – Comparison between brake specific fuel consumptions obtained with an engine in good and in poor condition, for the same fuel-air ratio**

1. Allowable variations in carburetor metering.
2. Test equipment accuracies.
3. Engine operating characteristcs with variation in mixture strength.
4. Engine operating differences with respect to altitude.
5. Inherent characteristics of the particular carburetor model.
6. Differences between reference carburetor metering tendencies and production carburetor metering tendencies.

Obviously, there is one man who should know all of the factors involved, and he is the carburetor engineer who has conducted the experimental engine tests on the particular carburetor model. Therefore, it is necessary, when analyzing flight data, to make sure that the results are checked by the carburetor engineer before too many conclusions are drawn.

The allowable variations in carburetor metering have been covered previously, and it has been found that the actual total variation in fuel-air ratio, compared to the standard of engine guarantees, might be $^{+8}_{-4}\%$ in the rich-mixture range, and $^{+10}_{-4}\%$ in the lean-mixture range, with an expansion of these tolerances of 1% in each direction for operation above 16,000 ft. *Actually, production carburetors will perform much better than the above tolerances would indicate, partly because of some built-in compensating errors, and partly because the above tolerances represent extreme, and not average, performance.*

The question of test equipment accuracies has been thoroughly considered previously, and it is believed that no review is necessary on this item.

The effect of variation in mixture strength on engine operating conditions depends largely on the fuel-flow range being considered. From observations of engine characteristics during constant-power, constant-speed mixture control curves, it is evident that in the higher fuel-flow range, the manifold pressure remains essentially constant for a wide variation in fuel-air ratio. The automatic rich setting is constantly in the high fuel-flow range. In the lower fuel-flow range, a smaller change in fuel-air ratio results in a large change in manifold pressure, with a very slight change in fuel consumption. Considering a typical 14-cyl engine, we therefore note that the percentage changes shown in Table 1 will occur on a mixture control curve at 1900 rpm and 745 bhp on propeller load.

Table I – Effect of Variation in Fuel-Flow Range on Engine Operating Conditions

		Approximate Change, %
High Fuel-Flow Range		
Fuel-Flow Range	371 to 460 lb per hr.	24
Manifold Pressure Change	27.24 to 27.44 in. Hg.	1
Change in Specific Fuel Consumption	0.50 to 0.62 lb per bhp-hr.	24
Fuel-Air Ratio Change	0.073 to 0.089	22
Low Fuel-Flow Range		
Fuel-Flow Range	330 to 371 lb per hr.	12
Manifold Pressure Change	28.74 to 27.24 in. Hg.	5
Change in Specific Fuel Consumption	0.44 to 0.495 lb per bhp-hr.	12
Fuel-Air Ratio Change	0.0625 to 0.073	17

Analyzing the table, it is evident that a proportional change of fuel-air ratio in the low and high fuel-flow ranges will cause a disproportionate change in manifold pressure and specific fuel consumption in the ranges shown. This condition undoubtedly makes the analysis of carburetion data extremely difficult, unless a suitable background of testing has been acquired on the particular engine being flight tested.

Another factor which is not often considered in analyzing carburetor data obtained in flight is the fact that the specific air consumption decreases approximately 2% in 20,000 ft for the same power output. This change would cause a decrease in metering differential for a constant power climb of 1 to 2%, which would affect the apparent mixture as judged by brake specific fuel consumption.

On most experimental airplanes, the carburetor is usually of a new design, incorporating supposed improvements which might change the inherent characteristics of the carburetor model. It is not usually feasible to acquaint the personnel running the flight test with the inherent char-

acteristics of the particular carburetor model, since these data can only be gained by test-stand experience over a long period of time. It would appear that this item could only be evaluated with regard to the flight data by the carburetor engineer.

Many times we discover unsatisfactory trends in the metering characteristics of reference or experimental carburetors which are corrected by the time the model is in production. It is extremely important that these tendencies be considered during the analysis of carburetor flight data, as sometimes these trends account for more variation in fuel-air ratio than the combined effects of the airscoop and other variables previously mentioned. In order to take care of all these variables, and properly to analyze flight data with a minimum of confusion, a chart, similar to that shown in Fig. 12, has been developed for plotting carburetor metering data.

You will note that this chart is plotted against three variables, namely: airflow, engine speed, and specific manifold pressure, which allows easy comparison of test-stand and flight-test data as well as airbox data. On this chart is shown the performance of the engine used during the determination of the reference carburetor setting. Limits are established for fuel-air ratio, specific fuel consumption and fuel flow, but these limits only hold for an engine of the same mechanical efficiency as the experimental engine. Likewise, these limits can only be expected to hold for test data obtained with equipment identical to that used in determining the reference-carburetor setting. Therefore, it is necessary when using this curve to check flight data, to expand the limits to include all the variables discussed previously. If the data obtained during the flight test are plotted on this chart, it will be apparent if the engine used for the flight test is putting out the same brake horsepower for the same airflow and manifold pressure as the experimental engine. Likewise, unsatisfactory test data will be immediately noticeable since the fuel flow obtained for a given metering suction will not correlate with the curves obtained on the reference carburetor.

To illustrate the use of the chart, two typical flight points "*A*" and "*B*" have been plotted. Analyzing point "*A*" data, we see that the specific fuel consumption, brake horsepower, and fuel flow agree with the reference carburetor data, but that the metering suction and manifold pressure are not in agreement with the reference curves. If this point were plotted against brake horsepower, and analyzed on the basis of specific fuel consumption, the analysis could only show that the data was satisfactory. However, by using the other data shown on the curve, we would consider point "*A*" as an unsatisfactory point because:

1. The manifold pressure is high, indicating an inefficient engine; but the metering suction is low, indicating an efficient engine, and this relation cannot be, since both the manifold pressure and the metering suction should definitely correlate with the engine efficiency.

2. The fuel flow plots within the limits for a low metering suction, and with most carburetors this relation cannot be true if the carburetor is metering properly.

Analyzing the "*B*" data, it would appear from the brake specific fuel consumption curve that the carburetor is metering too lean. However, on considering the additional data, we note that the metering suction, fuel flow, and manifold pressure are low, while the horsepower is in agreement with the reference data. From this analysis, we would say that the point was satisfactory and indicated good carburetor metering for the following reasons:

1. The engine is more efficient than the engine used to obtain the reference-carburetor data, since this engine puts out the expected horsepower with a low manifold pressure.

2. The manifold pressure and metering suction are both low, indicating the flight engine is drawing less air than the reference engine.

3. The brake specific fuel consumption is low, indicating that the fuel-air ratio for the flight engine is slightly lower than that obtained with the reference engine, because of the increased efficiency of the flight engine.

Of course, the above analysis only holds provided we are sure that all gages, flowmeters and the torquemeter have been properly calibrated. Otherwise the data shown in both points "*A*" and "*B*" could be analyzed to show instrumentation errors of considerable magnitude.

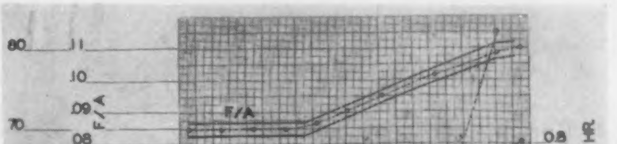
Summary

1. The approach to perfection in carburetor metering is based on the degree of skill applied to measurement.

2. To expect present carburetors to meter to closer limits than $\pm 5\%$ over the entire engine operating range is wishful thinking.

3. Present methods of checking carburetion in flight do not necessarily determine the complete functional performance of the carburetor, since the measurements taken reflect, to a large extent, the engine efficiency, as well as the efficiency of the personnel and equipment used.

Considering the variables involved, the quality and functional performance of present carburetors are considered satisfactory from a practical operating standpoint, and with further cooperation of the aircraft, carburetor, and engine manufacturers, should continue to improve in a normal manner.



■ Fig. 12 – Propeller-load, reference-carburetor data – sample curve

Terminal Handling of Air Cargo

continued from page 288

the aircraft is not available and probably will not be until large cargo carriers designed around such a system are in use. Mobile overhead hoists, when designed for high lift and high capacity, become too ponderous to do the fast and accurate job required, as is also true of the portable, powered ramp.

Probably the best and maybe the only hoisting device that can do the all-around job is the fork lift truck. Unless the ramp can be kept clean at all times, the common, small factory lift truck with small smooth wheels and low clearances cannot be used. Models with larger treaded wheels are available that have been doing good work in factories, warehouses, and yards for years. With a little further development they can do the job required. Improvements must and can be made in maximum overall height to permit maneuvering under wings, visibility for the operator, lifting and lowering speeds, steering effort, and convenience of controls.

The fork lift truck can pick up any packaged piece of cargo that is pallet mounted, any loaded skid platform, or any cargo train cart and elevate it to cargo door level and hold it for unloading into the aircraft or place it inside the aircraft for further handling by the cargo stowing crew. Hand-lift trucks inside the aircraft can pick up and move the pallet-mounted load or the skid platforms into position. If the aircraft is large enough, the cargo train cart can be moved inside the cargo hold to the point of unloading for stowing. With proper ship design, the pallet-mounted package or the skid platform with the load already strapped to it can be bolted to the cargo floor and would require no further lashing.

■ Requirements of Unloading

The reverse of all of these operations applies equally well when unloading.

Communications between the pilot of the incoming plane and the terminal control tower, advising of the pilot's load and his arrival time; between the control tower and the pilot, directing him to his berth after landing; from the loading supervisor to the general loading crew; from the ground crew back to the load dispatch headquarters, and in general between all concerned, whether they are in the cockpit of the aircraft, in the cargo hold, on a truck on the ramp, in the warehouse, or in flight is of utmost importance for accurate and speedy handling. No other means of communication meets all needs as does the two-way radio, with possibly a combination of radio and loudspeaker call system running a close second. Good communications will make good coordination of movements and efforts possible but cannot be of greatest value unless it is coordinated with a good system of dispatch of written material. It is just as essential that the delivery of the load manifests, the cargo records, the waybills, the receipts, and the clearances move fast from load and ship dispatchers to ramp loading crews and back again as it is from these that they get their verbal instructions and can make their verbal reports. The pneumatic tube system of distribution probably has no equal for speed and flexibility, and if properly planned, it can meet the requirements of the cargo terminal.

Good lighting of the ramp, the aircraft, and the load handling areas at night is necessary to make possible fast movement of cargo handling equipment without collision with other equipment, with men, or with aircraft and to assist in the identifying and handling of cargo in the direct vicinity of the airplane. Concentration of the aircraft near the warehouse, and loading and unloading from the warehouse side of the aircraft will ease this lighting problem.

In this paper I have tried to avoid any discussion of the relation between the cargo handling problem and the size and type of aircraft, and the size and weight of cargo to be handled. It is important that the terminal and its facilities be so designed and selected that any type of aircraft can be berthed, unloaded, loaded, and dispatched with uniform and equal efficiency and that any size or weight or class of cargo that might be placed on board these aircraft be handled in a uniform manner with equal efficiency and with a common, flexible, type of equipment.

New Windshield Developments

continued from page 293

In further relating the tests to practical variations in impact conditions which would be realized in actual collisions, an investigation was made of the effect of the location of the point of impact on the face of the windshield panel, and of the effect of variations in angle of impact which might be obtained with the bird moving transversely to the line of flight. It was learned that the penetration velocity of the glass vinyl type of windshield construction is decreased approximately 25% when the impact occurs near the outer end of the panel, as compared to impact at the center or inboard end. It further was found that a similar decrease in penetration velocity of about 25% is obtained for impact at a lateral angle of 11 deg from the normal and a vertical angle of 17 deg from the normal. However, such an angle of impact would only be obtained when the speed of the airplane was reduced to approximately 110 mph. This is roughly 60% of the cruising speed.

The tests conducted so far mainly have involved investigations of the effect of various types of panel construction and mounting and the effect of other pertinent variables. As yet no attempt has been made to combine the more promising features in a single windshield which would possess a maximum of impact resistance. They have indicated, however, that the glass vinyl type of laminated windshield with extended plastic edges bolted into an adequately reinforced windshield frame provides practically 100% protection for the DC-3 airplane against collisions with birds weighing up to 4 lb, and that a high degree of protection practically may be provided against collisions with birds up to 20 lb in weight. Furthermore, such protection is possible for all anticipated impact conditions and for all ambient temperatures to be expected. The net increase in weight as compared to existing installations of double-glazed windshields and including reinforcement of the windshield supporting structure will be approximately 15 lb. The optical characteristics are believed acceptable.

The Use of PETROLEUM PRODUCTS in AIRCRAFT

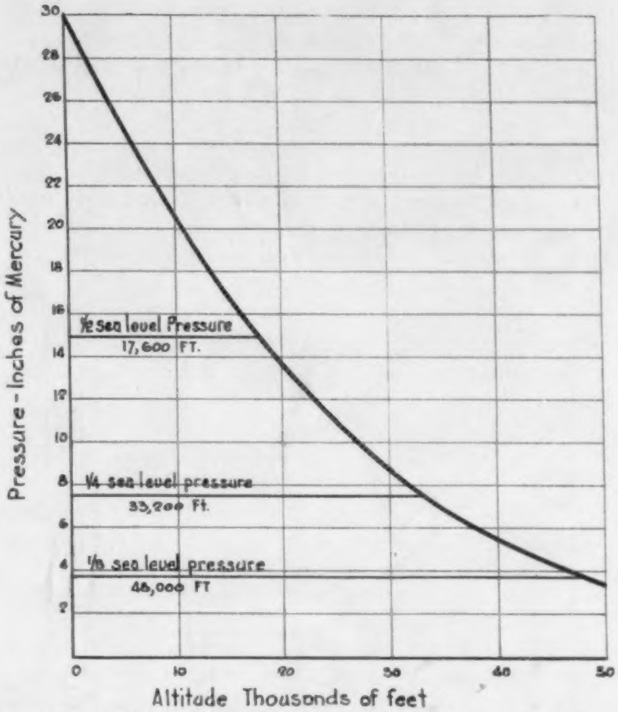
by FRANK D. KLEIN
Standard Oil Co. of N. J.

IT seems well to begin with an examination of the atmospheric conditions which contribute to the reasons why aircraft products must frequently exceed in quality the products which are satisfactory for ground applications.

Aircraft operate under a wide range of atmospheric pressures varying from sea level to less than one-fourth of sea level. Fig. 1 shows the variation in pressure with altitude. It is seen that atmospheric pressure at 17,600-ft altitude is half of that at sea level; at 33,200 ft it is only one-fourth; and at 48,000-ft altitude it is only one-eighth of that at sea level. These pressure variations influence chiefly the vapor-locking tendencies of the fuel system. Other factors bear consideration, however. For example, under conditions of thorough contact and agitation, air is soluble in aircraft hydraulic oil to an appreciable extent at atmospheric pressure. When the pressure on hydraulic oil, so saturated with air at normal sea-level pressure, has been reduced to 75 mm (equivalent to an altitude of 10,750 ft) the volume of air liberated is substantial, in comparison to the volume of the oil itself. Although the major part of the hydraulic system of aircraft remains under high pressure, designers must consider the air which is liberated at low pressure in the supply reservoir. Also of current importance is the release at low atmospheric pressures of air dissolved in the engine lubricant, which in some cases may contribute to the problem of oil tank foaming.

Temperature variations have a marked effect on the properties of lubricants, and atmospheric temperature limits encountered in aircraft are, therefore, of interest. As is

[This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Okla., Oct. 22, 1942.]



■ Fig. 1 – Chart showing the variation in pressure with altitude (for standard atmosphere)

DISCUSSING the meteorological conditions under which aircraft operate, and so the conditions that rule the specifications of fuels and lubricants, the author points out that aircraft are used under a wide range of atmospheric pressures. Pressure variations influence chiefly the vapor-locking tendencies of the fuel system. Solubility of the air in the hydraulic oil and in the aircraft-engine lubricating oil also introduces problems, under wide fluctuations of atmospheric pressure.

Greatest need in engine lubricating oil development is improved stability, with consequent reduction in engine deposits. Detergent-type oils are effective in improving engine cleanliness and reducing ring sticking.

Most grease lubrication can be accomplished with only two grades – a high-temperature grease and a low-temperature grease. The low-temperature grease is used chiefly for intermittently operated units that are never subject to high temperatures. The high-temperature grease is used for continuously operated units, which are subject to low temperatures for only short periods and which normally are kept warm.

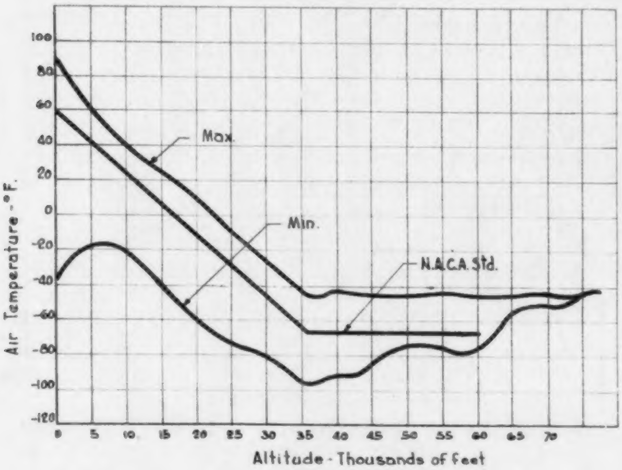
Corrosion preventives for aircraft fall into two general classes: One for the protection of exterior surfaces of parts, and the other for interior protection of engines. They consist generally of a rust-inhibiting base incorporated in a liquid or solid vehicle, depending on the desired method of application and the type of protective film.

■ ■ ■

THE AUTHOR: FRANK D. KLEIN (M '42) was retired from active military service with the rank of captain in 1939. He had been a member of the Army Air Forces since his graduation from Massachusetts Institute of Technology in 1925. He spent two years at The Glenn L. Martin Co. as assistant chief of laboratories. In November, 1941, he joined Standard Oil Co. of N. J., to handle technical aviation matters.

well known, ground temperatures frequently exceed 110 F, not only in deserts, but in some of the hotter airports in this country. Aircraft standing in the sun at these high temperatures often become much hotter. At the low-temperature end of the scale, many of our northern areas experience ground temperatures as low as —20 to —40 F. Until recently, is was generally believed that the stratosphere, in which temperature ceases to fall with increasing altitude, began at about 36,000-ft altitude and had a constant temperature of —67 F. It is now known that the tropopause, which marks the lower level of the stratosphere, varies in height above the earth's surface, being higher near the equator than near the poles. Even in a given locality it varies somewhat in height from day to day. The temperature of the stratosphere varies inversely as the height of its lower surface above the earth, which accounts for stratosphere temperatures of approximately —50 F over the poles as compared with —120 F over the equator.

AiResearch Mfg. Co. has tabulated some very interesting data on temperatures at various altitudes and localities, based upon U. S. Weather Bureau reports for the year 1940. A portion of its data has been reproduced here. First, consider the temperatures recorded at a northern station, as shown in Fig. 2. The lowest temperature en-

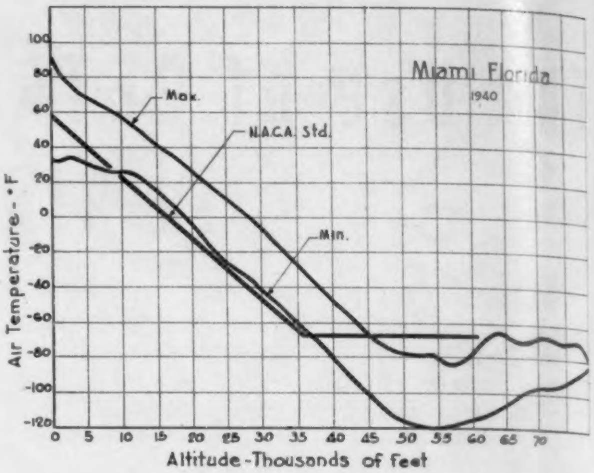


■ Fig. 2 – Variation in temperature with altitude at a northern station

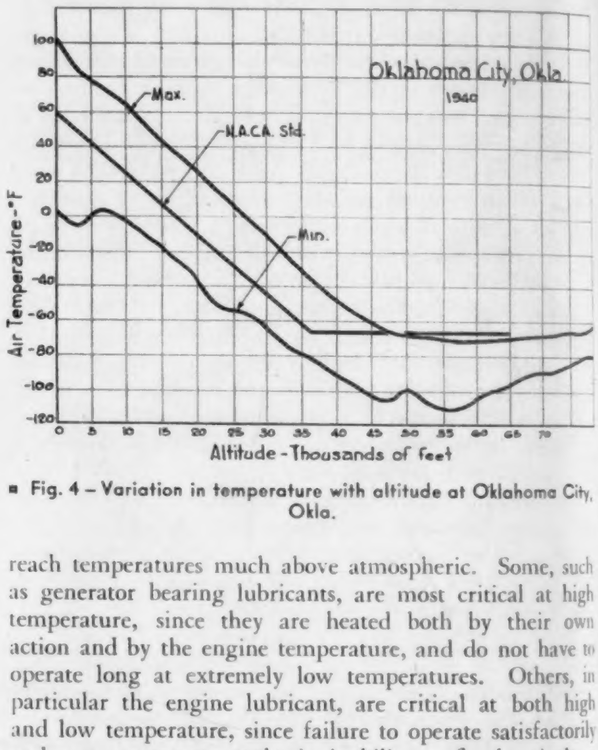
countered on the ground was —39 F and the highest temperature 88 F. The lowest temperature at altitude was —97 F, which occurred at 36,000 ft. For comparison, consider how a warm climate such as at Miami, Fla., fares. From Fig. 3 it is seen that the highest temperature encountered was 92 F, only a few degrees above that shown in Fig. 2. The lowest temperature on the ground was 30 F, which does not compare too unfavorably with their advertising. However, it is surprising to note that a temperature of —120 F was encountered at 53,000-ft altitude. Fig. 4 shows the conditions at Oklahoma City.

In Fig. 5, the altitude and temperature of the tropopause are plotted against latitude, based upon data from various localities. It is clear that from the equator to the north pole the tropopause lowers and its temperature increases.

Having described the atmosphere in which aircraft must fly, the requirements and limitations of petroleum products for use in aircraft are better understood. Some of these products, such as control bearing lubricants, are most critical at the low temperatures discussed, since they never



■ Fig. 3 – Variation in temperature with altitude at Miami, Fla.

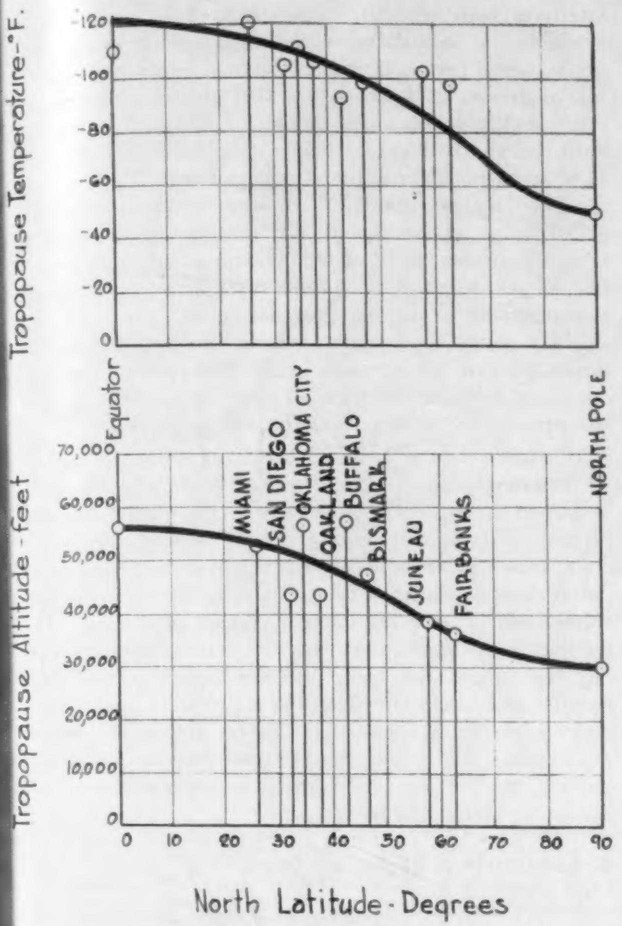


■ Fig. 4 – Variation in temperature with altitude at Oklahoma City, Okla.

reach temperatures much above atmospheric. Some, such as generator bearing lubricants, are most critical at high temperature, since they are heated both by their own action and by the engine temperature, and do not have to operate long at extremely low temperatures. Others, in particular the engine lubricant, are critical at both high and low temperature, since failure to operate satisfactorily at low temperature results in inability to fly the airplane, and failure at high temperature causes engine difficulties or increased maintenance.

Little can be reported on recent developments of engine oils. The greatest need is improved stability, with consequent reduction in engine deposits. Detergents similar to those found so necessary for high-speed diesel engines have proved effective in improving aircraft-engine cleanliness and reducing piston-ring sticking. Cylinder wear is naturally of prime concern in any lubricant investigation, and it seems well to observe, in view of some opinions expressed in this respect, that present evidence does not indicate that cylinder wear is necessarily increased with the use of detergent-type lubricants. As to the use of antioxidants, separately or in conjunction with detergents, it would appear that addition agents of this type ultimately may contribute importantly to the stability requirements of lubricants for high-output engines.

The oil dilution system is in general use on aircraft



▪ Fig. 5 – Chart showing the temperature and altitude of the tropopause plotted against latitude

which must be started quickly at very low temperatures. A small portion of the engine oil is diluted with gasoline upon shutting down the engine, so that easy cranking will be obtained for later starting. In addition to reducing cranking torque, this dilution lowers the pour point of different types of oil to a varying degree. For greatest effectiveness with the oil dilution system, the engine oil should be reduced rapidly in pour point when diluted with small percentages of aviation gasoline.

Maximum efforts are being exerted toward the greatly increased production of 100-octane fuel of required full-power performance characteristics, and the further improvement of antiknock value above 100 octane. Many new alkylation and catalytic cracking plants are in operation and many more are under construction to augment the supply of high-octane fuels. Moderate quantities of special addition agents are being used as a means of increasing full-power performance. This has necessitated the development of fuel system materials resistant to many fuels.

With aircraft ceiling continually being raised, vapor-lock problems require serious consideration. Under present-day high-altitude-flight conditions, air solubility in gasoline is of much importance and actually tends to offset the effect of decreased front end volatility under conditions approaching vapor lock. Many newly developed protective measures, such as pre-flight cooling of the fuel on the ground and centrifugal and float controlled vapor separators, appear to be feasible.

Engines in some types of light planes, it should be observed, have had to suffer unavoidably by change in fuel quality necessitated by increased requirements for high-octane fuels. Until recently, unleaded 73-octane number fuel and 80-octane number fuel containing not over ½ cc of tetraethyl lead per gal were in general use. Under current Government restrictions on maximum base stock octane number, these fuels are no longer available. Present 73-octane number fuels contain about ½ cc of lead per gal, and 80-octane number fuel contains about 1¾ cc of lead per gal. Some of the light engines designed to operate on unleaded 73-octane number fuel are giving satisfactory service with either the present 73- or the 80-octane number grade, because of the use of materials resistant to the corrosive action of the products of combustion of leaded gasoline. Exhaust valves, valve seats, and exhaust disposal systems are the most critical items in this regard. Engines not so protected are experiencing increased maintenance difficulties due to the high tetraethyl lead content. The solution to this problem appears to depend upon the ability of the engine manufacturers, in the face of critical material shortages, to supply necessary parts of corrosion-resistant material, since there seems to be no hope of a change in the Government restriction on aviation fuels for some time. All of the larger engines operate satisfactorily with fuel containing up to at least 4 cc of tetraethyl lead per gal.

Two grades of hydraulic oil are now standard for aircraft hydraulic systems. The medium grade is in general use, and is satisfactory down to temperatures as low as 0 F to —20 F, depending upon the particular equipment. For lower temperatures, the light grade has been necessary, and with newly designed pumps now available, it can be used for most of the hydraulic equipment under all climatic conditions.

A hydraulic oil, to give excellent performance under all conditions, should have a viscosity about equal to the present light grade at —40 F and about equal to the present medium grade at 130 F, with other properties, including rubber swelling, pumpability, and leakage, about equal to the present medium grade. A specification for such a product has already been issued. Special additives to improve oiliness are available, but have not proved necessary and are not permitted by latest specifications.

There has been some indication of a need for rust-preventive type of hydraulic oil for use in unusually corrosive atmospheres, such as near sea water. While such products are available, it is not yet known whether the need will be serious enough to justify service use.

Marked improvements have been made recently in certain specialty oils for aircraft use. It has been found desirable to improve low-temperature properties and stability of instrument oil, and to add agents to improve oiliness and impart rust-preventive qualities. Similar improvements have been made for machine gun oil.

For a limited number of applications, such as lubrication of remotely driven propeller reduction gears, an extreme-pressure oil is used. It is possible that improved low-temperature characteristics will be needed because of the increase in cold-weather flying.

Aircraft greases probably are receiving more attention at this time than any other type of petroleum lubricant for aircraft. In any discussion of greases, it must be borne in mind that, in general, oil provides better lubrication than does grease, provided that it can, without leakage, be maintained in contact with the surfaces to be lubricated.

However, besides having greater load-carrying ability than straight mineral oils, greases possess the marked advantage of changing less in consistency with temperature variations than do oils, thus making it easier to keep them in their place through wide temperature fluctuations.

The activity now involving aircraft greases probably is due chiefly to the following:

1. Need for improved low-temperature characteristics because of increased flying in cold weather and at high altitudes.

2. Development of improved laboratory tests to evaluate such vital characteristics as oxidation stability, bleeding, and low-temperature torque.

3. Development of sealed "lubricated-for-life" bearings.

4. Development of improved greases suitable for greater temperature range.

5. Desirability of simplifying the lubrication problem by reducing the number of greases required, both by improving the suitable performance range of greases and by consolidation of lubricant recommendations so as to increase the number of units for which available greases are suitable.

An ideal grease would be one which could be used for all units requiring grease lubrication for the entire temperature range encountered. Unfortunately, no single grease is known at present which will meet these requirements. One type of grease has just become standardized which satisfactorily lubricates most of the aircraft and accessory units subject mainly to low temperatures. This covers almost all intermittently operated units, such as control bearings, retracting mechanisms, as well as some continuously operated units, such as magneto gears.

The low-temperature grease in general use at the present time is a water-resistant product formulated with oil of low viscosity and pour point. Specifications limit such characteristics as bleeding, oxidation, and low-temperature torque. It should be emphasized that low-temperature torque cannot be judged by the consistency or appearance of the grease, since it is based upon the properties of the base oil, together with the characteristics of the soap and the method of formulation. For example, an oil in the fluid state might readily have higher torque than a grease of stiff consistency.

Units which operate a large part of the time appreciably above normal atmospheric temperature require the use of high-temperature grease. These generally include such items as continuously operated motor and generator bearings, and also wheel bearings, since they often get very hot due to the action of the brakes.

Various types of high-temperature grease are now in general use, but standardization of a specification is expected in the near future, which should provide for almost all grease lubrication which cannot be accomplished with the present low-temperature grease.

While many units, such as retracting gear boxes and other screw mechanisms, which were formerly believed to require extreme-pressure lubricant are now operating satisfactorily with low-temperature grease, there has been a limited but definite need for an EP grease of good low-temperature characteristics. A product that meets these requirements is a modification of the present low-temperature grease containing an EP additive. Unfortunate though it seems for simplification, there also appears to be some need for a semifluid grease of good low-temperature characteristics for use in some gear mechanisms in which channeling of the standard grease at low temperature seriously interferes with adequate lubrication. Such a product is available, but manufacturers should determine by test that the standard grease is not satisfactory before resorting to special greases. Similarly, now that greases are becoming more standardized, units should be designed to operate with the standard grades whenever possible, avoiding designs requiring EP or other special lubricants, and providing good sealing. In the past, some mechanisms which galled unless operated with EP lubricant were modified to operate satisfactorily with conventional grease by changing the gear material or, in some cases, by merely changing heat-treatment to increase gear hardness.

Many points on aircraft, such as hinges and joints, require "oil can" rather than grease lubrication. For best operation, a low-temperature oil combining lubricating and rust-preventive properties may be used.

To assist personnel who service many models of aircraft in correctly lubricating each model, lubrication diagrams similar to those used at automotive service stations are very helpful. Fig. 6, representing several models of light aircraft, shows an example of how this is done. Each lubrication point is identified by a square, indicating grease gun application, or a circle, indicating hand application. The numbers inside the squares and circles are symbols identifying the proper lubricant, and the figures beside these squares and circles indicate the number of flight hours between servicing operations. Larger airplanes with several engines, wing flaps, engine cowl flaps, and many retracting mechanisms are naturally more complicated, but similar diagrams can be prepared.

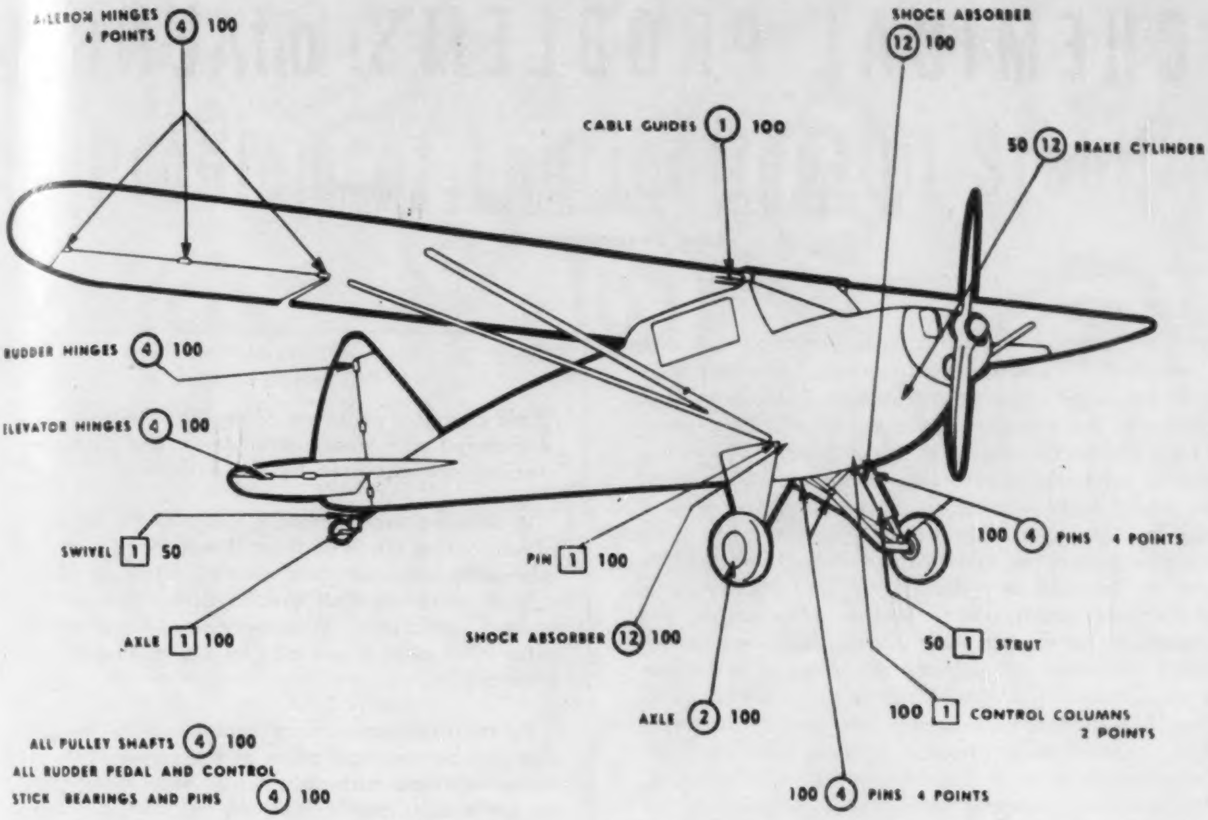
Lubrication of Accessories

In spite of the desire to simplify greasing, many accessories now require special lubricants. Aircraft manufacturers and operators would be benefited greatly if as many accessories as possible could be lubricated with the greases being standardized for the airplane itself. In many cases this is feasible, but in some cases it is not possible to avoid specialized lubricants. Inertia starters and Prestone pumps of liquid-cooled engines now use special lubricants, but standardized greases might later prove satsfactory. Space does not permit a discussion of the lubrication of the innumerable other accessories used in aircraft.

Various types of anti-seize products are available for the threaded fittings of different classes of equipment, such as spark plugs, fuel and oil lines, hydraulic lines, oxygen lines, and other fittings.

Prevention of corrosion in conjunction with lubrication has already been mentioned in connection with certain specific lubricants, so the discussion will not be repeated. Corrosion prevention during storage and shipment, as applied to aircraft, falls into two distinct classes: the protection of exterior surfaces of parts, and the interior protection of aircraft engines.

As regards exterior parts protection, no single corrosion preventive will meet all requirements. Primary consideration must be given to whether the parts are to be stored indoors, outside in open sheds, or outdoors exposed to the elements. These factors largely govern the consistency required of the corrosion preventive. For example, outdoor exposure would necessitate a relatively hard protective film, which would normally be applied in the hot state, whereas indoor exposure would usually be satisfactory with a thin film applied by dipping or spraying at room temperature,



■ Fig. 6 – Lubrication diagram – light airplanes

SYMBOL	TYPE OF LUBRICANT	RECOMMENDATION
1	HIGH PRESSURE GREASE	
2	HIGH TEMPERATURE GREASE	
4	LIGHT LUBRICATING OIL	
12	HYDRAULIC OIL	
	ENGINE LUBRICATING OIL	

□ GREASE GUN APPLICATION ○ HAND APPLICATION

THE NUMBER INSIDE THE SQUARE OR CIRCLE IS THE LUBRICANT SYMBOL, AND THE FIGURES OUTSIDE THE SQUARE OR CIRCLE INDICATE THE NUMBER OF FLIGHT HOURS BETWEEN SERVICING OPERATIONS.

and readily wiped off when desired. Corrosion preventives generally consist of a rust-inhibiting base incorporated into a liquid or solid vehicle, depending upon the desired method of application and type of protective film needed for the particular storage conditions involved.

Concerning the interior protection of aircraft engines, there are two special requirements not applicable to other corrosion preventives. First is the ability to neutralize hydrobromic acid, traces of which are formed in engine interiors due to condensation of the products of combustion of leaded gasoline. Second is the requirement that engines be operable on the corrosion preventive compound without detrimental effects, since treatment procedure involves operation of engines on the compound in place of engine oil for the last 15 min of running before shutdown for storage or shipment. For this reason, the compound must have properties similar to those of engine oil itself, together with rust-preventive characteristics.

■ Humidifier Life

Among the many specified requirements of all corrosion preventives, probably the most significant one is the humidifier life test. Equipment and conditions have not yet been universally standardized, but usually involve a chamber in which specimens are mounted and subjected to controlled temperature and humidity conditions. The specimens are examined periodically. The number of hours before initial appearance of rust spots is considered the humidifier life of the particular corrosion preventive. In such tests, the life of conventional mineral oils of moderate viscosity is usually about 1 to 2 hr. A life of well over 100 hr is desirable for any corrosion preventive. Estimated performance based on results of humidifier tests must be confirmed by service tests.

In closing, it seems apropos to mention that, although other petroleum products, such as some synthetic rubbers, are of value in aircraft, they are subjects in themselves.

■ Acknowledgments

The author gratefully acknowledges the courtesy of AiResearch Mfg. Co., Los Angeles, Calif., in authorizing publication of weather data from its Report PC-20-R entitled "The Variation of Air Temperature with Altitude."

CHEMICAL PROBLEMS of ENGINE LU

by R. G. LARSEN, F. A. ARMFIELD, and G. M. WHITNEY
Shell Development Co.

THE chemical problems of engine lubrication associated with wear, detergency, and oxidation stability are discussed by the authors.

A detailed examination is made of the stability of lubricating oils in engines. It was found that the oil rapidly becomes contaminated with solid oil-insoluble materials that greatly affect its stability toward oxidation: In some cases, the stability after 1000 miles of use may be 1% of that of the fresh oil.

By the addition of pure compounds to the oil, and also by the separation of the components of the oil-insoluble materials, it was determined that the metal salts, particularly the halogen salts, are the effective catalytic materials. Two possible methods for combating these catalysts are mentioned by the authors.

A few experiments are also reported indicating a possible usefulness of these engine catalysts in laboratory oxidation tests.

■ ■ ■

THE AUTHORS: R. G. LARSEN started his career on the staff of the Department of Electrical Engineering at Massachusetts Institute of Technology, where for two years he studied problems of breakdown of dielectric oils. He joined the staff of the Shell Development Co. in 1937 and became head of the lubricating oil department in 1940. F. A. ARMFIELD has been engaged in research on solvent extraction problems since he joined the Shell Development Co. in 1935 following his graduation from the University of California with a bachelor's degree in chemistry. More recently he has been working on methods for measuring and controlling deterioration of lubricating oils. He is co-author with R. G. Larsen of several papers on lubricating oil oxidation and catalysis. G. M. WHITNEY was with the Hercules Powder Co. during World War I, supervising the manufacture of nitroglycerine. Afterward he was connected with the Raven Oil & Refining Co. in charge of production and manufacturing, and with the Midwest Refining Co. (Standard Oil Co. of Indiana), where he was plant chief chemist and then assistant general foreman of manufacturing. From 1923 to 1927 he was a research chemist with Standard at Casper, Wyo., when he joined the staff of Shell Oil Co. working successively as chemist at Wilmington, head of distilling at Wilmington, and chief chemist at Dominguez. In 1937 he transferred to research on fuels and lubricants with the Shell Development Co. His experience with oils has resulted in the development of techniques which are especially applicable to the analysis of dark oils, tars, and asphalts.

OILS are used in engines to reduce friction and carry away heat. To fulfill these functions, it is simply necessary to use an oil of proper viscosity and volatility. If an engine were not a machine designed to convert the potential energy of matter into useful work by means of chemical processes involving blast-furnace temperatures, oxidizing gases under high pressure, high concentrations of soot, aldehydes, acid vapors, liquid acids, and steam, and if the moving parts were not subjected to sudden tremendous increases in the load they must carry, any viscous liquid would probably permit carefree and efficient operation. But the facts are that engines wear out because moving metal parts are permitted to touch and abrade away, or because they are attacked by corrosive vapors; rings stick because soot and fuel-and-lubricant-deterioration products are forced into ring grooves under pressure, lacquers form which decrease piston clearance and interfere with heat conduction, sludge forms and interferes with oil flow, and bearings are corroded by acids formed by fuels and lubricants. These are the problems the chemist finds when he studies the problems of engine lubrication.

Frequently, as in the case of ringsticking in diesels, the lubricant is not the chief cause of the difficulties. The fuel may often be at fault, or poor mechanical design may not provide an opportunity for the lubricant to function properly.[1] Nevertheless, it may be that the lubricant can remedy the situation by exerting an action other than that of cooling and reducing friction. Thus, if it exerts a detergent action, preventing soot or other solids from coagulating and adhering to the metal, ringsticking may be reduced or eliminated. Since such an action is always beneficial, the ability to keep an engine clean may be added to the necessary functions of a lubricant. Studies made in these laboratories on this subject have already been reported.[2]

The role of the lubricant in reducing friction and wear is not a simple one. Wear is partly a mechanical or metallurgical problem involving engine design, strength and hardness of metals, and provision for proper lubricant film thickness. It is also a chemical problem involving the ability of the lubricant to form coatings on metal surfaces to protect them from destruction even if the lubricant is temporarily squeezed out. Much of the wear of engine surfaces is the result of rusting which occurs under cold-start conditions; thus the rust preventive properties of the lubricant become important.

From the complex field of the chemical behavior of lubricants, that of stability of oils toward oxidation has

[This paper was presented at a meeting of the Northern California Section of the SAE, Oakland, Calif., Nov. 10, 1942.]

[1] See SAE Transactions, Vol. 48, May, 1941, pp. 165-173: "Engine Design Versus Engine Lubrication," by R. J. S. Pigott.

[2] See *Industrial and Engineering Chemistry*, Analytical Edition, Vol. 15, Feb. 15, 1943, pp. 91-95: "Lubricating Oil Detergency," by S. K. Talley and R. G. Larsen.

NE LUBRICATION:
the Problem of Lubricating Oil Stability

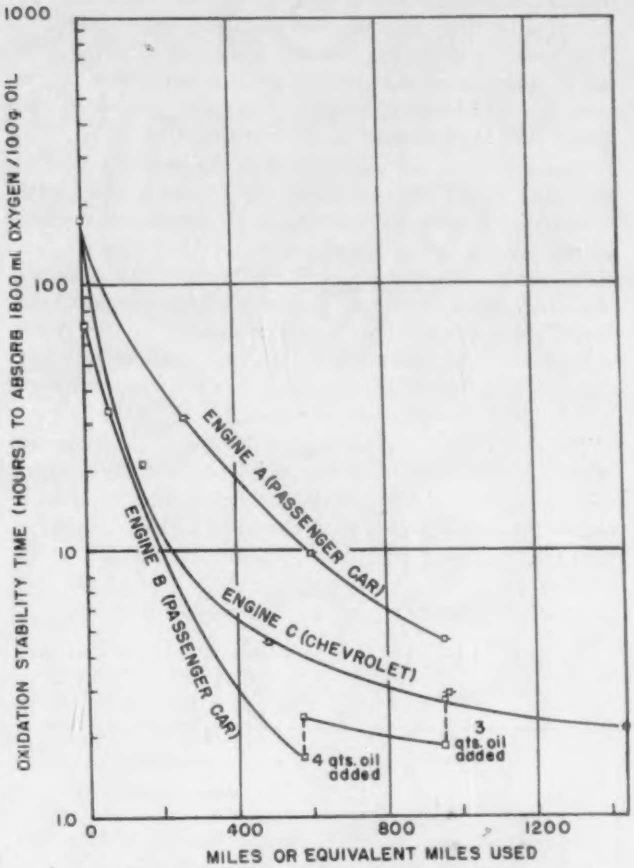
been chosen as the general subject of the present paper. The formation of lacquers and sludges as well as corrosion of bearings is usually attributed to attack of the oil by oxygen. This view is substantiated by the fact that lacquers and sludges contain large amounts of combined oxygen, usually close to 20%. Furthermore, if oils remain neutral and do not contain oxygenated acidic compounds, bearing corrosion does not occur. Ring-groove deposits may also contain large amounts of combined oxygen. Thus, oxidation stability may become a problem the chemist must understand and control in order to provide better lubricants.

Doubts are sometimes raised as to the practical significance of oxidation in actual operation, for in many cases apparently satisfactory performance is obtained over considerable periods with relatively unstable oils. Traces of incipient oxidation may even be somewhat desirable for improving "oiliness" characteristics. However, it may be pointed out that there is a potential danger of sudden failure if deterioration should proceed too far. Very frequently, if unstable oils do not have one bad effect, they have another. Thus detergent-type oils, even when badly oxidized, keep an engine clean, but, by virtue of this very action, expose bearing surfaces to corrosive attack by acids. On the other hand, unstable oils which form protective coatings over the bearings generally lay down sludge or lacquer deposits elsewhere in the engine, which interfere with normal operation. In any case, the most innocuous deterioration products (oil-soluble, noncorrosive) give rise to a viscosity increase and may thus transform a light-grade lubricant into a heavy one for which the engine is not designed. Obviously, if an oil possesses suitable characteristics when it leaves the refinery or is charged into an engine, it is desirable that these not be altered.

Much has been said and written about the relation between the stability of lubricating oils and their composition. Pennsylvania oils were "stable" while the so-called naphthenic and asphaltic base oils could be improved by refining to remove "unstable" aromatics. A short time ago, some of us presented a discussion of the oxidation stability of high-molecular-weight pure hydrocarbons.[3] It was shown that "aromatics" are not unstable; some, particularly the polynuclear ones, are very stable. It was concluded that stability of an oil is determined by small amounts of natural inhibitors rather than by the hydrocarbon components of the oil. A recent paper by von Fuchs and Diamond[4] indicates that these antioxidants are probably associated with the natural aromaticity of lubricating oils.

[3] See *Industrial and Engineering Chemistry*, Vol. 34, February, 1942, pp. 183-193: "Oxidation Characteristics of Pure Hydrocarbons," by R. G. Larsen, R. E. Thorpe, and F. A. Armfield.

[4] See *Industrial and Engineering Chemistry*, Vol. 34, August, 1942, pp. 927-937: "Oxidation Characteristics of Lubricating Oils: Relation between Stability and Chemical Composition," by G. H. von Fuchs and H. Diamond.



■ Fig. 1 – Oxidation stability of crankcase oils at 150 C as a function of miles used

■ Stability of Oils in Engines

These general studies afford considerable information regarding oil stability, refining methods, and the like, but a gap still exists in our understanding of the stability of an oil in an engine. It is the particular intent of this paper to present some of our most recent findings on this subject.

Stability in an engine is usually judged by the change an oil undergoes when subjected to engine operation under controlled conditions; increase in viscosity, neutralization number, or some other variable usually is used as a criterion of the change. This does not, however, give the chemist an understanding of the fundamental nature of the deterioration. It was decided that the desired information could best be obtained by periodically removing samples of oil from an engine and studying oxidation stability in glass apparatus. Thus, if any changes occurred in the intrinsic properties of the lubricant, they would likely become evident by a change in stability.

In order to have a standard starting point, the passenger-car engines used in this study were first drained free of

the old oil, "flushed" by running for 200 miles on the test oil, and finally refilled with the test oil, and the test was begun. The laboratory engines used were completely dismantled, cleaned in Bendix cleaner and Oakite, reassembled, and tested directly. In all cases, the periodic oil samples were removed through a stopcock in the oil pressure line while the engine was still running and after the line had been flushed by removing several hundred milliliters of oil before the sample was taken. The excess oil thus removed was returned to the crankcase to minimize the amount of make-up oil required. Although three different oils were studied in several engines, the data of Fig. 1, which shows the decrease in stability of a lubricating oil of commercial grade when used in two passenger cars and in one laboratory engine (Chevrolet run under proposed ASTM conditions[5]), are representative.

Stability of the oil was judged by the time required for the oil to absorb 1800 ml of oxygen per 100 g of oil, when oxidized in a glass apparatus, without addition of catalysts or purification of the oil in any way. The apparatus was described in an earlier report.[3] Under these conditions, the oil is sufficiently deteriorated to have a neutralization number of about 4, and a saponification number of 16. A survey of over 100 passenger-car used-oil analyses indicated that this amount of deterioration is about the maximum which will be found in service.

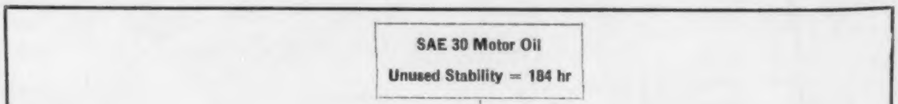
The data of Fig. 1 show that in the case of passenger-car engine B, the stability decreased to one-half the original value in 10 min of idling, presumably the time required to ensure thorough mixing with the small amount of oil left after the "flushing" procedure. In all cases, stability of the oil dropped rapidly, so that after a short time (equivalent to less than 1000 miles) of use the stability was as low as 1% of that of the fresh oil. Engine B was in poor mechanical condition and caused a greater decrease in stability than Engine A. The addition of 4 qt of fresh make-up oil (total crankcase capacity 7 qt) led to but little improvement, indicating that a minimum stability had been reached. The fact that the oil used in the passenger-car engines under the mild conditions of city driving became as reactive as that used in the Chevrolet under greatly accelerated oxidizing conditions indicates the generality of the phenomenon.

These results immediately raise the question as to the importance of the stability of fresh oils. Is it worthwhile for the refiner to produce a stable oil if the engine so mistreats it that it becomes very reactive when used a few miles? Perhaps the laboratory rating of stability is meaningless when it is made in glass (as in the old Indiana, the British Air Ministry, and the Sligh tests) with no engine metals (iron, copper, or lead) present. However, the used oil from the engine was also oxidized in glass apparatus in the absence of engine metals except for the small amount of abraded iron it might contain, so acceleration by addition of catalytically active metals may likewise be meaningless. The most logical explanation is that the oil "picked up" in the engine a catalyst which increases the rate of reaction between oil and oxygen. Since these catalysts are obviously not simple engine metals, it becomes essential to identify them to understand the problem of stability.

In order to discover the nature of this catalyst present in the used oil, the final drainings from Engine B were examined in detail as shown by the flow sheet of Fig. 2. Removal of the oil-insoluble portion by filtration through a very fine glass filter greatly improved stability, indicating that the insoluble solids are very active catalytically. When the fuel residues were removed from the filtered oil by steam distillation, stability was improved very little. Removal of all the oil-soluble oxidation products and metal components by treatment of the filtered oil with hydrochloric acid and clay restored the stability of the oil almost to its original value. Since the filtered oil contained oil-soluble salts of lead, copper, and iron, these results indicate that the oil-insoluble materials contribute more to reactivity than the small amount of oil-soluble metals, although the latter are generally considered to be the most important factor in catalysis of oil oxidation.

The solid oil-insoluble material was highly oxidized, as indicated by a saponification number of 283 mg KOH per g. It was of course very high in ash, which contained principally compounds of lead. The metals and metal-containing compounds were removed from this material by Soxhlet extraction with constant-boiling hydrochloric acid. The resulting residue still contained the highly oxidized material but

[5] See ASTM Standards on Petroleum Products and Lubricants, 1942, pp. 28-42: "Proposed Method of Test for Oxidation Characteristics of Heavy-Duty Crankcase Oils."



■ Fig. 2 – Flow chart showing the separation of the catalyst from the used oil

was reduced greatly in ash content, that remaining being almost entirely compounds of lead. Contrasted to the original high-ash crankcase solids, which were very active catalytically, the low-ash solids actually exerted a slight inhibiting action. The latter effect was confirmed by oxidizing the same oil in glass apparatus, removing the oil-insoluble sludge (Saponification No. 240) and adding it to the fresh oil, when it too acted as an inhibitor.

The evidence at hand thus indicates that the finely dispersed oil-insoluble solids (which we have chosen to call "crankcase catalysts") occurring in crankcase oils are responsible for the catalyzed activity of such oils. This confirms the accelerating effect of similar materials previously noted by Davis.[6] In order to study broadly their activity and composition, a number of samples of crankcase catalysts were obtained from various sources, with results as summarized in Table 1. Three composite samples were obtained from service-station drainings (from two localities) and probably represent the drainings from over 200 cars. In addition, deposits were obtained from aviation engines and from a motor car which was run with nonleaded and with leaded fuels. In most cases, the solids were obtained by centrifuging the warm oil (at about 90 C) in a Sharples Super Centrifuge (20,000 × gravity). As would be expected, the ash content of all the samples was high; the use of leaded gasoline led to an increase in ash and an accompanying increase in lead and halogen content. It is of particular interest that the catalytic activity of all these solids was high except when nonleaded fuel was used.

Table 1 – Analysis and Activity of Crankcase Catalysts from Various Sources

Source:—		Service Station Drainings			1000-Mile Drainings Dodge Car			
Analysis*	From Oil Used 950 Miles in Engine B (from Fig. 1)	Sample No. 1 from 25 gal taken in California	Sample No. 2 from 30 gal taken in California	Sample No. 3 from 8 gal taken in Illinois	Nonleaded Fuel	Leaded Fuel	Crankpin Deposit Pratt-Whitney Wasp 500 hr	Crankcheek Deposit Wright C.U.E.
Ash	35.8	47.2	46.0	....	22.4	48.1	75.3	69.1
Lead	19.8	27.4	36.4	27.9	2.9	24.2	47.0	47.0
Iron	4.5	5.4	5.8	3.0	10.66	10.69	7.66	18.27
Copper	0.12	0.34	0.07	0.11	0.08	0.17	0.3	0.80
Sulfur	....	4.3	....	4.3	4.5	3.5	0.56	0.36
Halogen**	....	7.4	10.91	7.1	0	10.1	16.0	12.0
Bromine	....	4.5	6.74	3.7	0	6.2	....	11.0
Chlorine	....	1.3	1.85	1.5	0	1.7	....	0.45
Carbon	....	30.6	22.1	36.3	....	....	15.2	17.6
Hydrogen	....	2.6	....	3.4	....	....	0.84	1.05
Oxidation Time***	2.3	4.5	3.0, 2.8	2.15	25.0	3.0	3.36	8.05

* Percentage basis.
** Calculated as bromine.
*** Time (hr) to absorb 1800 ml of O_2 per g of oil when 0.5%w catalyst is added to SAE 30 motor oil, oxidized in glass apparatus at 150 C. The dried catalyst was powdered and passed through a 325-mesh screen before being added to the oil.

■ Source and Nature of Crankcase Catalyst

The next step in the study was to identify the source and nature of the catalytic substances contained in the crankcase catalysts. The first clue is, of course, that the crankcase catalyst formed when nonleaded fuel is used is much lower in catalytic activity. Leaded fuel forms large quantities of lead salts, principally the oxide and halide, some of which find their way into the oil. The halide is formed by interaction of lead compounds with the added alkyl-halide components of the fuel during the combustion process. Likewise, sulfur compounds in the fuel may react to form lead sulfide or sulfate. The halogen compounds (probably the acids) may also attack piston, cylinder, and valve surfaces to produce the halides of iron and aluminum. Rusting and abrasion of the ferrous surfaces will also produce the various oxides of iron. Besides these, the sulfides and sulfates of iron may be present. The presence of copper in the engine leads to the possible formation of the halides, oxides, sulfides, and sulfates of copper. The total amount of copper present is so small, however, that unless it is considerably more active than iron, it would probably be over-shadowed by the latter. Besides the salts listed above, it is also possible that a small amount of finely abraded iron is present as the free metal. To study the activity of the substances described above, the action of the pure salts when added to fresh oil was tested, with results as summarized in Table 2. To find a material representative of abraded iron, piston-ring filings were obtained and screened to obtain a fine powder. For comparison the results are also given for the addition of the naphthenic-acid salts of lead, copper, and iron. These salts are all oil-soluble and, it is believed, represent the form in which the metals may exert their maximum catalytic activity. The inactive materials were added in 0.5%w concentration; the active ones were added in the same concentration as they occur in 0.5%w of crankcase catalyst.

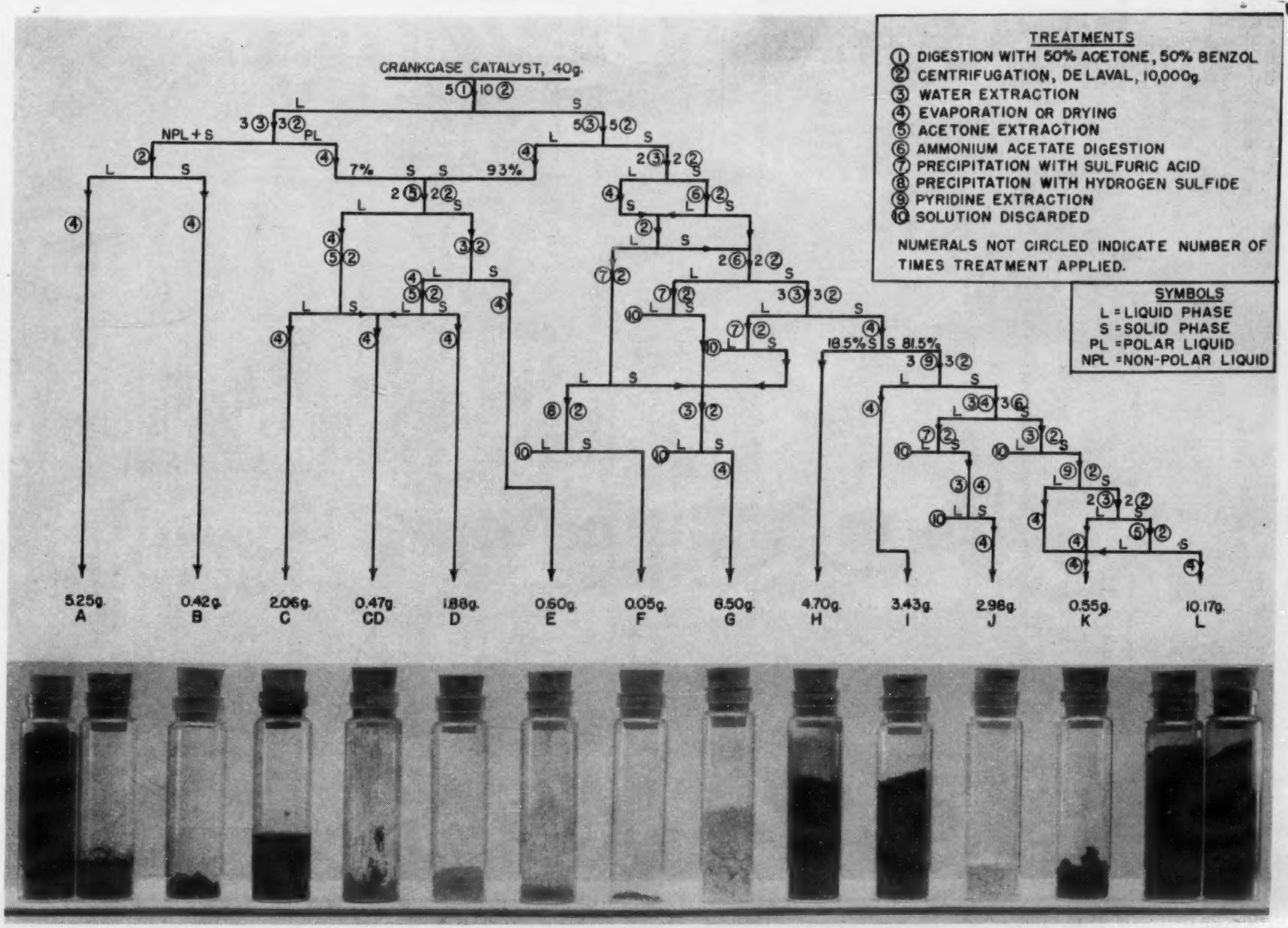
[6] ASTM Bulletin, December, 1941, pp. 13-17: "The Examination of Used Engine Crankcase Oil," by L. L. Davis.

Table 2 – Catalytic Activity of Various Inorganic Materials When Added to SAE 30 Motor Oil

Added Material	Amount Added, %w	Time to Absorb 1800 ml O_2 per 100 g oil at 150 C, Hr
None		213
Crankcase Catalyst composite sample 1	0.5[2,3]	4.5
Lead Oxide	0.5	150
Lead Sulfate	0.5	212
Lead Sulfide	0.5	200
Lead Bromide	0.5	173
Lead Naphthenate	0.5	42[1]
Ferric Oxide	0.5	63
Ferrous Oxide	0.5	20.5
Ferric Sulfate	0.5	198
Ferrous Sulphate	0.5	233
Ferrous Sulphide	0.5	38.5
Ferric Nitrate	0.5	28.0
Ferric Chloride	0.078[2]	1.50
Ferrous Bromide	0.104[2]	2.55
Ferric Naphthenate	...[2]	1.3[1]
Iron, reduced powder	0.027[2]	71.5
Iron, Piston-ring filings >325 mesh	0.027[2]	115
Cupric Chloride	0.0036[3]	11.2
Cupric Naphthenate	......[3]	8[1]
Aluminum Chloride	0.5	136

[1] Interpolated Values.
[2] Iron content, equal to 270 p.p.m.
[3] Copper content, equal to 17 p.p.m.

The data show that the inorganic lead salts are generally very inactive, only the oxide and bromide having appreciable catalytic effect, which is still much lower than that of an equivalent quantity of crankcase catalyst.



■ Fig. 3 – Flow chart of the separation of the crankcase catalyst

The salts of iron are all effective, except the sulfate. The halides are extremely active (being quite comparable to the naphthenates) and even more so than an equivalent quantity of iron in the crankcase catalyst. It is thus apparent that iron halides must be considered as important components of crankcase solids. It is doubtful, however, if all the iron is present as halide, since oxides, sulfates, and small quantities of free iron are present. This is at least partly indicated by the fact that a substantial portion of the catalyst is attracted to a magnet.

Copper chloride is also a very active substance and must be considered as possibly contributing to activity of crankcase solids.

The relative inertness of the free metals even when finely divided is shown by the results with the piston-ring filings. The reduced iron powder is more reactive but probably is not representative of crankcase material. Aluminum chloride is likewise of little activity.

Since the halides of copper and iron could result only from the use of leaded fuel, it is interesting to compare the data in Table 2 for the Dodge car run with leaded and nonleaded fuel. Although the iron contents of the two used-oil solids are identical, the activity is increased almost ten-fold when halogen is also present.

To check the significance of these results, an attempt was made to separate the crankcase catalyst into its components with as little alteration as possible in their chemical state. This is an extremely difficult problem since the organic materials present form protective "envelopes" around the active particles which hinder the action of solvents. Also, the extreme complexity of the mixture renders it improbable that a perfectly homogeneous substance could be obtained by the methods employed. Nevertheless, it was possible to separate the catalyst into significantly different fractions and largely to confirm the conclusions based on the addition of pure substances to the fresh oil.

The diagrammatic summary in Fig. 3 indicates in detail the procedure employed in the separations (using service-station drainings Sample No. 1); the size of the samples gives an idea of the relative amounts of each fraction present. The more important fractions are briefly discussed here. It may be mentioned that satisfactory separations between solid and liquid fractions could not be made by filtration due to the extreme fineness of the solids; centrifugal separation was therefore necessary.

The *organic portion* (sample *A*) is obtained by solution in a mixture of 50-50 benzene-acetone, which dissolves also some iron and lead salts. To free it from these latter substances, it is washed with water several times. As seen by the analysis given in Table 3, this sample is low in ash and has little activity.

The *iron halide fraction* (sample *C*) is obtained by first digesting the residue from the acetone-benzol extraction with water, evaporating the aqueous extracts to dryness, and then dissolving the iron (and copper) salts in acetone; the lead halide is not soluble in this solvent. The analytical data of Table 3 indicate that this sample is relatively free of lead and contains but a small amount of copper. This fraction is very active catalytically, giving in 0.1% concentration an activity greater than that of the original catalyst. The actual separation of this fraction is believed to be of considerable significance.

The *lead halide fraction* (sample *D*) is the water-soluble acetone-insoluble portion remaining after removal of the iron halides. The analysis in Table 3 indicates that it is approximately 80% pure; its catalytic activity is somewhat greater than one would expect from the study of pure lead bromide. However, if the contaminants are copper and iron halides, they may contribute to the activity.

Table 3 – Analysis and Catalytic Activity of the Fractions Separated from the Crankcase Catalyst

Sample	Original	A	C	D	E	F	G	H	Dehalogenated Original
Characteristic Composition	Total Deposit	Organic Portion	Iron Halide	Lead Halide	Mixed C & D	Copper Sulfide	Lead Sulfate	Residue	....
Ash, %	47.2	2.70	....	....	....	....	....	43.75	....
Lead, %	27.4	1.09	0.12	45.7	21.92	....	66.5	15.33	33.9
Iron, %	5.4	1.34	5.53	0.24	29.7	....	....	3.23	7.4
Copper, %	0.34	0.063	0.105	0.072	0	16.0	....	0.416	0.4
Total Halogen*, %	7.4	1.0	36.8	38.3	3.4	....	....	0.72	4.8
Sulfur, %	4.3	....	....	....	....	....	....	1.89	7.6
Carbon, %	30.6	....	....	....	....	....	....	30.8	....
Hydrogen, %	2.6	....	....	....	....	....	....	3.6	....
Sulfate, %	9.85	....	....	0.7	5.6	....	....	5.85	....
Oxidation Time,** hr	4.5	105	3.22***	55	....	....	172	13	38.5

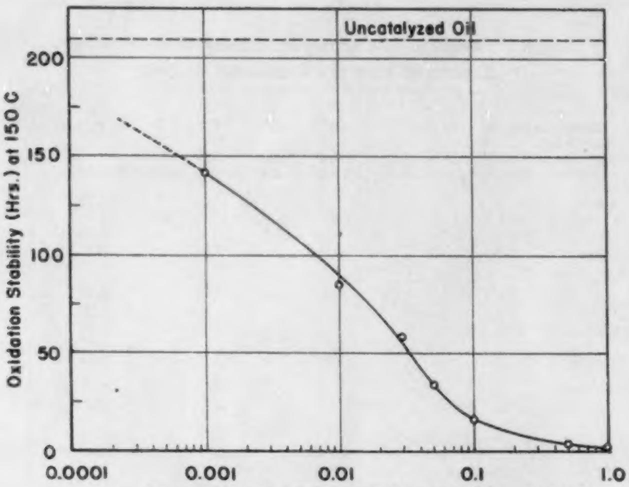
* Halogen calculated as bromine.
** Time for 100 g of oil to absorb 1800 ml of oxygen when fraction is added in 0.5%w concentration. Oxidation at 150 C.
*** Fraction added in 0.1% concentration.

The *lead sulfate fraction* (sample *G*). Tests showed that lead sulfate could be obtained from the residue remaining after extraction with cold water by digesting it with hot water. This method is very slow, so instead, an extraction was made with ammonium acetate solution and the lead reprecipitated by addition of sulfuric acid. This procedure admittedly involves a conversion of the original material but, since lead sulfate was demonstrated to be present, would not alter the ultimate conclusions. Considerable quantities of lead sulfate were thus separated which, in agreement with earlier findings, was quite inactive.

After separation of the various fractions as described, the residue (sample *H*) was still active and contained lead, copper, and iron somewhat changed in ratio only, an indication of the fact that probably only part of the material was acted upon by the solvents used. At this stage, pyridine extraction was tried, but, although its solvent power for organic materials is greater than that of benzene and acetone, it also dissolved more of the metallic components and therefore did not produce significant fractions. The original crankcase catalyst was also heated to very high temperatures in a stream of inert gas in an attempt to decompose or sublime the halides from the mixture. Decomposition and sublimation did occur as indicated by tests on the exit gases, but not all the catalytic activity was eliminated. This work did serve, however, to give considerable confidence to the conclusions derived from the study of the pure salts themselves.

■ Effect of Crankcase Catalyst

An interesting fact shown in Fig. 1 is that the oil became reactive soon after it was added to an engine; the data of Fig. 4 provide an explanation. Thus, when as little as 0.001% of crankcase catalyst is added to a fresh oil, stability is reduced to about two-thirds of its original value. As little as 0.1% reduces stability to less than 8% of the original, while 0.5% reduces stability to about 2%. The oil-insoluble content of most used oils is greater than 0.1%, and therefore approaches the maximum catalytic activity or "catalytic saturation." For most practical cases, the oil



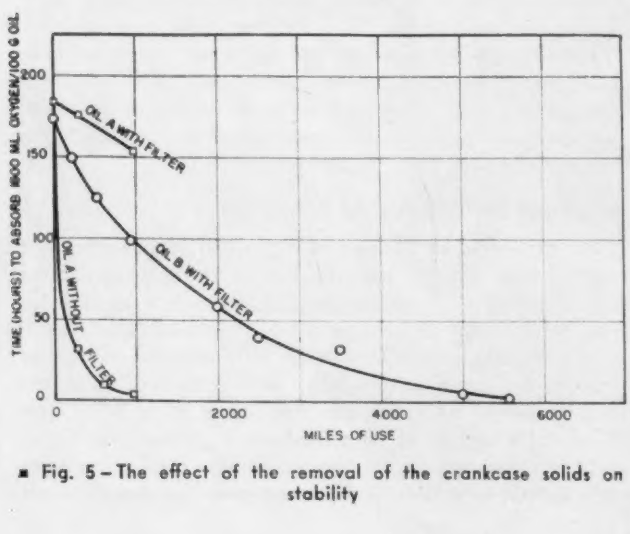
■ Fig. 4 – Effect of the concentration of the crankcase catalyst on the stability of SAE 30 oil

is subjected to these effects almost from the beginning of its use in an engine.

■ Elimination of Catalytic Influences

Two methods may be employed to overcome the catalytic effect of the solids which accumulate in crankcase oils. The first is the use of chemicals (anti-catalysts) which passivate or deactivate the catalysts. One such material has been studied in the laboratory, and it almost completely eliminates the catalytic influences of crankcase solids on used oils. It still remains to be seen if, when used in an engine, it will continue to function properly. Undoubtedly, this or some other lubricating oil additive will be found to accomplish the purpose.

The second method of solving the problem is to insert an efficient filter in the engine oil system. From what has been shown, it is obvious that, to be effective, the filter must completely remove the suspended solids. To test this possibility, a Mercury car was run in two tests of 1000 miles each on the same oil, first without and then with a filter. The differences in stability of the oil are clearly seen in Fig. 5. Another test was made in which the car was run without oil change for 5000 miles; after the engine was overhauled. During this period, stability gradually decreased even though the oil remained free of suspended



■ Fig. 5 – The effect of the removal of the crankcase solids on stability

solids, probably because of the gradual accumulation of oil-soluble metal salts. These data then indicate for this particular oil and car the maximum benefit that can be obtained with a filter and indicate also the limitation on oil-change periods. The beneficial effect of the filter is obvious and should receive thoughtful consideration from the standpoint of maintaining oil stability.

■ Significance of Crankcase Catalysts

At present, there is no complete answer to the significance of crankcase catalysts to engine operation. Indeed, it is not expected that an unequivocal answer will be found. Under some conditions of engine operation, it is quite likely that almost all of the oil deterioration occurs in the power section, where temperatures are so high that the oil reaching that section (at least the part that does not vaporize) is extensively oxidized regardless of stability. In other cases, considerable deterioration does occur in the crankcase. Or again, a detergent may cause an unstable oil to leave the engine clean even though the oil is badly oxidized. Bearings may not be corroded by acids formed by unstable oils, since such oils may form a protective coating on the bearing surface. Therefore, the effects of oxidation stability may be obscured by other engine factors.

Since leaded fuels produce the most active catalysts, an easy method for checking the importance of catalysis would be to run an engine under carefully controlled conditions with and without leaded fuel. The data presented in Table 4 for a pair of such experiments indicate that considerably more oxidation occurred when leaded fuels were used. This is by no means to be taken as a generality, however, since the high sump temperatures of the test engine were conducive to the catalyst exerting its maximum effect. Also, lead compounds form detergents, and in some cases may lead to cleaner engines, and perhaps may modify corrosive characteristics of the oil by removing protective films from bearings.

Table 4 – Deterioration of Oils in The Lauson Engine Jacket Temperature, 100 C – Sump Temperature, 140 C 40-Hr Drain Period

	Fuel			
	Nonleaded		Leaded	
Oil	Neutralization No.	Saponification No.	Neutralization No.	Saponification No.
C	3.75	10.2	6.13	48.8
D	1.15	7.9	2.61	14.2

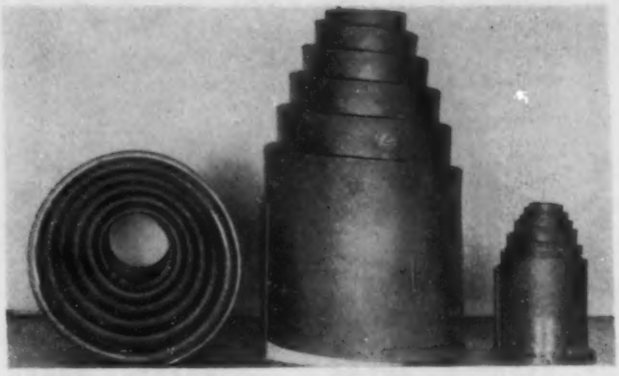
■ Crankcase Catalysts in Laboratory Tests

Since engine lubricants are subject to the effect of crankcase catalysts from the time they are first added to the engine, "stability" should perhaps be judged in the presence of these catalysts rather than arbitrarily chosen amounts and types of metal or metal-salt catalysts. While it is not proposed now to urge the general use of crankcase catalyst, it would nevertheless seem a logical choice to make. The data of Fig. 6 summarize the studies that have been made in this direction. The stability of fresh oils, as judged by the time required for oxidation to a given extent in the laboratory in the presence of 0.5% crankcase catalyst, is plotted against the degree of oxidation they underwent when tested in the engine. Deterioration in the engine is judged by the increase in saponification number, which has been shown to represent about 20% of oxygen actually

turn to page 343

A DESIGN METHOD for VOLUTE SPRINGS

by H. O. FUCHS
General Motors Corp.



■ Fig. 1 – Views of volute springs

THE volute spring, long used in railroad work, is now being applied to automotive equipment. The more stringent demands on space and performance in the automotive field have made refinements in materials, design, and manufacture necessary.

Mr. Fuchs here presents a discussion of the design phase of the problem.

Gradually increasing spring rate (stiffness) and unequal stress distribution along the blade are the chief features of volute springs. Apart from scale factors, Mr. Fuchs says, rate increase and stress distribution depend only on the ratio of smallest to largest free coil radii and on the variation of free helix angles from coil to coil.

Use of nondimensional charts simplifies design, enables a rapid survey to be made without calculations, and gives a correct picture of loads and stresses.

The importance of the presetting operation is emphasized by the author. Layout of coiling form and presetting bowl are a major design item, for which a rational procedure is proposed.

★ ★ ★

THE AUTHOR: DR. H. O. FUCHS (M '34) has been an engineer for General Motors Corp. since 1933 – first at Delco Products Division, Dayton, Ohio, and then in the central office – working mainly on dynamics and elasticity problems. He was a mechanics apprentice, automobile detail designer, lumber mill master mechanic, and patents engineer before 1933. Dr. Fuchs received a degree of Bachelier ès Lettres in 1924 from the University of Strasbourg; degree of mechanical engineer in 1929 and doctor of engineering in 1933 from the Karlsruhe Polytechnic Institute. A member of the Volute Spring Subcommittee of the Spring Committee of the SAE War Engineering Board, he received many suggestions for this paper through working on this subcommittee. He is also on the Leaf Spring Subcommittee of the SAE-W.E.B. Committee, and has published papers on shock absorbers, vibrations, model tests, volute springs.

VOLUTE springs have been used for many decades, mostly in railroad work. The automotive field, to which they have more recently been applied, makes more stringent demands on space and performance; refinements in materials, manufacture, application, and design have therefore become necessary.

The present paper is concerned mainly with the design phase of the problem; it owes much to the contacts and suggestions which the author has enjoyed in the Volute Spring Subcommittee of the Spring Committee of the SAE War Engineering Board, where problems and experiences are discussed by volute spring users and makers. The help given by the members of this committee is gratefully acknowledged. Thanks are due in special measure to the chairman, Bernhard Sterne, who, besides contributing many suggestions, took charge of the calculation and plotting of the type curves; to H. C. Keysor for well-considered criticism; and to the American Locomotive Co., American Steel Foundries, Eaton Mfg. Co., and Spring Perch Co., Inc., for furnishing data.

■ Main Features of the Volute

Two features of the volute make it interesting as a spring, and these same two features are the cause of many headaches: The load-deflection diagram is not a straight line and working stresses are higher than anything we allow in other coiled springs.

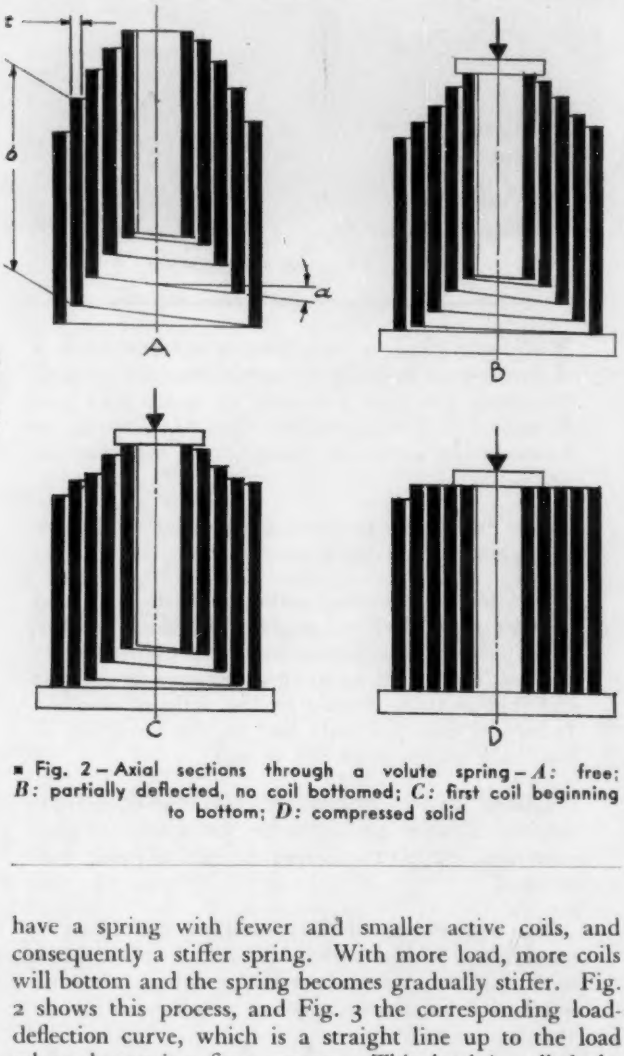
Fig. 1 shows views of typical volute springs. The spring is coiled from a rectangular blade of width b and thickness t. The radius of the coils gradually decreases from the large end to the small end. Each blade section has its own particular radius R and, in general, also its own free helix angle α. Thickness and width of the blade might also vary, but usually they are held constant for economy in manufacturing, except for the end coils which are tapered in thickness to give a circular outline and sheared in width to give flat seats on both ends.

When load is applied to the spring, the coils deflect and the helix angles decrease. If we consider an element of

[This paper was presented at a meeting of the Canadian Section of the SAE, Toronto, Ontario, March 17, 1943.]

the volute as equivalent to an element of a helical coil spring with the same radius R and helix angle α, we find[1, 2] that stresses and change of helix angle are proportional to the coil radius. Each element then acts like a spring with constant rate, and the sum of their deflections is proportional to the applied load.

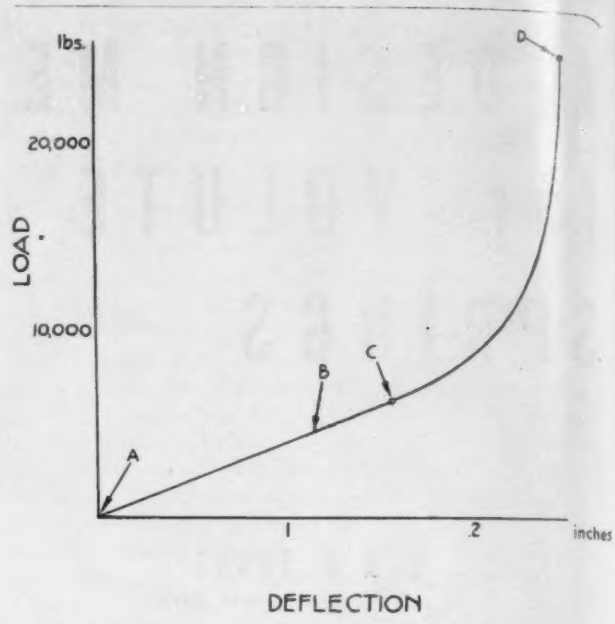
The largest coil will deflect most, as illustrated by spring B in Fig. 2, and at some load it will bottom. Then we



■ Fig. 2 – Axial sections through a volute spring – A: free; B: partially deflected, no coil bottomed; C: first coil beginning to bottom; D: compressed solid

have a spring with fewer and smaller active coils, and consequently a stiffer spring. With more load, more coils will bottom and the spring becomes gradually stiffer. Fig. 2 shows this process, and Fig. 3 the corresponding load-deflection curve, which is a straight line up to the load where bottoming first appears. This load is called the transition load, and the corresponding deflection, transition deflection.

Inspection of the spring and mathematical analysis show that a volute element does not act exactly like an element of a helical spring with equivalent radius:[3] However, the approximate assumption that a coil element acts like an element of a helical spring of the same cross-section, helix angle, and radius gives reasonably correct values for loads and deflections, and is so much simpler than a complete analysis, that what follows is entirely based on this approximation, which leads to the simple equations: Bottoming stress $= Gt\alpha/R$ at any cross-section with coil



■ Fig. 3 – Load-deflection curve of a volute spring – points A, B, C, and D correspond to the conditions of springs A, B, C, and D in Fig. 2

radius R and free helix angle α. Bottoming load $= G\eta_3 b t^3 \alpha / R^2$ for the load just sufficient to bottom the blade of width b, thickness t, and modulus G at a point with coil radius R and free helix angle α. η_3 is de Saint-Venant's coefficient $(1/\eta_3 \approx 3 + 2t/b)$. Coil radius R and free helix angle α vary along the spring blade. The manner in which they are distributed determines, to a large extent, the performance of the spring. The coil radius changes from the largest active radius R_l at the outer heel to the smallest active radius R_s at the inner heel. The change of radius from one coil to the next is constant, and is usually called the radial pitch.

The variation of free helix angle from coil to coil is not as obvious as the variation of coil radius, but it can easily be checked by measuring the height of the various coils above a base line. This variation has only recently been recognized as a valuable design element; older volute springs were coiled with a constant helix angle; the presetting or "bulldozing" then changed them into springs with variable helix angle by giving more permanent set to the very highly stressed inner coils than to the moderately stressed outer coils. No effort seems to have been made to produce a predetermined distribution of free helix angles. The helix angle will vary in some way from a high value α_l at the large end to a low value α_s at the small end. Plotting the helix angle α over coil radius R will give a curve, which for convenience can be approximated by a straight line. This assumption is usually justified by the actual distribution of helix angles.

If the shape of a spring – including the distribution of helix angles or the resulting pitches – is given, the load-deflection curve and the stresses can be calculated; the basic calculations are not very difficult, but quite tedious.

To relieve the designer of the tedium of working with long formulas, a method has been worked out which is very easy to apply, flexible enough to accommodate a wide range of possible volute springs, and gives a good insight into the features of various types of volute springs.

[1] See "Les Ressorts – Etude Complète et Méthode Rapide de Calcul," by Camille Reynal, 1938, Dunod, Paris.

[2] See SAE Transactions, Vol. 50, June, 1942, pp. 221-240: "Characteristics of the Volute Spring," by Bernhard Sterne.

[3] See ASME Transactions, Vol. 65, July, 1943, pp. 543-551: "Notes on Secondary Stresses in Volute Springs," by H. O. Fuchs.

This method is based on the fact that any two volute springs with the same ratios of R_s to R_l and of α_s to α_l will have proportionate load-deflection curves, regardless of blade section and number of coils. Cross-section and coil number will only change the scales of the curves. A number of load-deflection curves, which cover the range of possible ratios, have been calculated in advance. For a given blade section and number of coils, the scales for these curves are found by means of simple formulas and auxiliary charts which have also been calculated in advance to reduce the computing work as much as possible.

Efficiency and Stress Distribution

The same considerations apply to stresses. At this point, it might be useful to consider the efficiency of volute springs, which depends on the stress distribution. If that spring which gives the longest service with the least amount of steel is called the most efficient, it follows that stress distribution along the blade and the number of expected deflections of various amounts should be so balanced that the life expectancy is the same at any blade section.

Thus, if we expect many millions of small deflections and only a few large deflections, the bottoming stresses in the small coils may be chosen much higher than the static stress or the bottoming stress of the largest coil. This type of service may occur in certain suspension springs. Contrariwise, if the spring is always deflected fully at every stroke, best efficiency demands that the bottoming stress should be uniform along the blade. In the notation used here, the bottoming stress distribution depends on the ratios $q(q=R_s/R_l)$ and z, which defines the relative rate of change of the helix angle

$$z = \frac{\alpha_l - \alpha_s}{\alpha_l} \bigg/ \frac{R_l - R_s}{R_l}$$

For $z = 1$, the bottoming stress is uniform along the blade. (This corresponds to calculations shown by Holland.[4])

For $z = 0$, the free helix angle is uniform and the bottoming stresses are inversely proportional to the coil radius. (This corresponds to calculations shown by Sterne[2] and Wahl.[5])

For intermediate values of z, the conditions are between these extremes. Inspection of the type curves will show that low values of z and q give high change of spring rate. At the same time, they produce unequal distribution of stresses along the blade. Nonuniform stress distribution is the price which must be paid for the rising spring rate.

At high values of z, a change in q makes only a little difference in stress distribution. At low values of z, the stress distribution may become very unfavorable with small values of q. In one case, a spring was redesigned by leaving out some of the inner coils, which meant a higher q. Though the work was thereby distributed among fewer pounds of steel, the distribution was more nearly equal and the life was increased considerably.

Type Curves

The type curves are load-deflection diagrams which, in order to be useful for all springs, large and small, are given in terms of ratios instead of the usual units: pounds and inches.

The deflection f is expressed as a fraction of the total deflection h, and the load P is expressed as a multiple of the load M at ½ h deflection. M is always on the straight-line part of the curve.

The general type of the spring is defined by the ratios $q = R_s/R_l$, where R_s is the smallest and R_l is the largest free coil radius, and $z = \dfrac{1 - \alpha_s/\alpha_l}{1 - R_s/R_l}$, where α_s is the helix angle at R_s and α_l is the helix angle at R_l. This expression for z is derived by differentiating the ratio α/α_l with respect to the ratio R/R_l, and might be called the relative rate of change of helix angle.

A graphic representation of the meaning of various values of z is given in Fig. 4, where α/α_l is plotted against R/R_l.

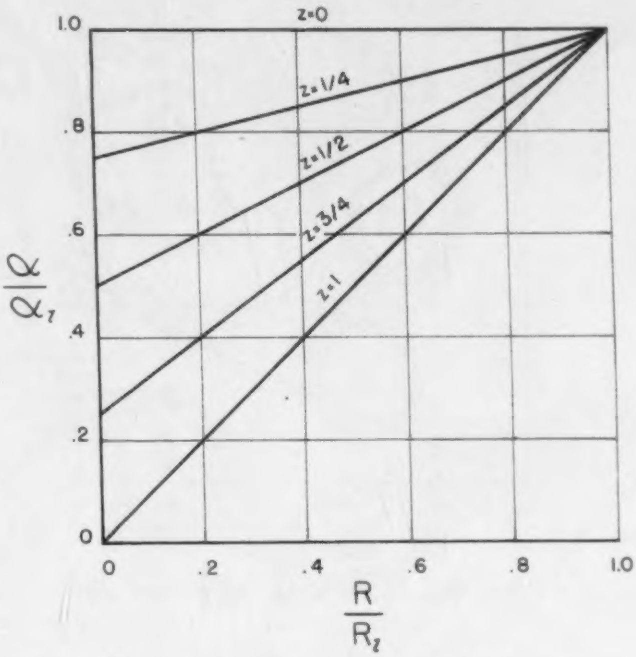


Fig. 4—Helix angle plotted as a function of coil radius, for various values of z

Type curves are given for:

$z = 0$ (Fig. 5)
$z = 1/4$ (Fig. 6)
$z = 1/2$ (Fig. 7)
$z = 3/4$ (Fig. 8)
$z = 1$ (Fig. 9)

In each of these figures, separate curves are drawn for q values of 0.3, 0.4, 0.5, 0.6, and 0.7. Curves for other values of z and q can be obtained by interpolation or by calculation from the formulas given in Appendix II. This will hardly be necessary because the difference from one type curve to the next is not much more than the difference of one spring to another made to the same drawing from a different heat of steel.

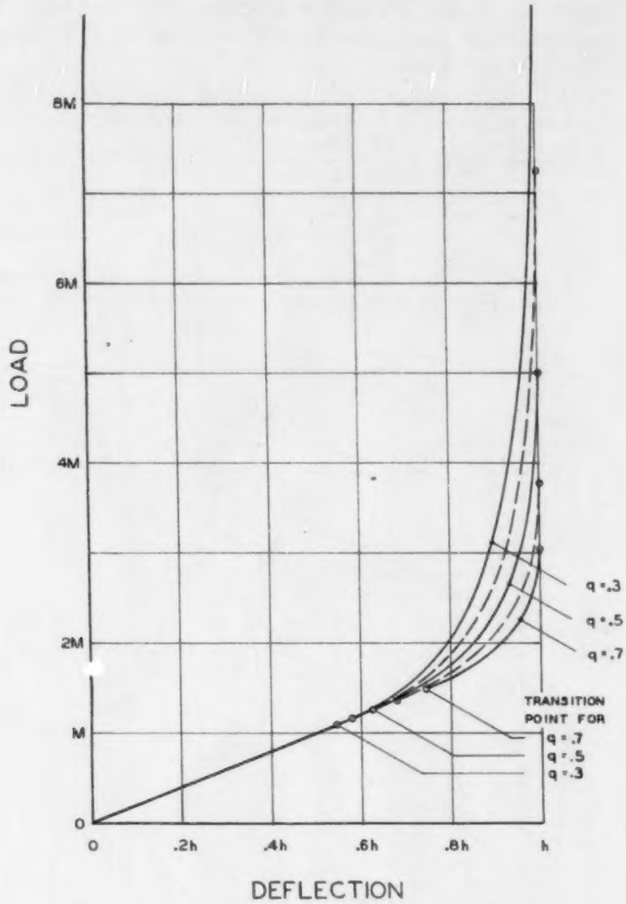
Individual load-deflection curves are obtained from the type curves by multiplying each abscissa (deflection) value by h and each ordinate (load) value by M.

M and h, as well as the initial rate S and the highest bottoming stress τ_s, can be found from the following formulas:

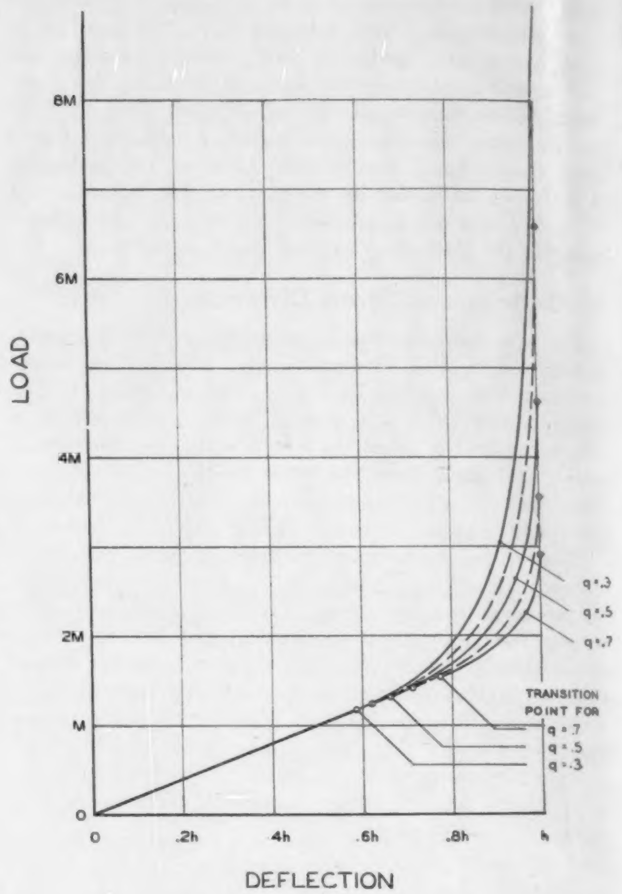
$$h = n\, R_l\, \alpha_s\, F$$

$$M = G\, \frac{b\, t^3}{R_l^2}\, \alpha_s\, Q\, F$$

[4] See ASME Transactions, Vol. 65, July, 1943, pp. 533-541: "Volute Spring Formulas," by C. J. Holland.
[5] See *Machine Design*, Vol. 15, February, 1943, pp. 103-107: "Simplifying Design of Volute Springs," by A. M. Wahl.



■ Fig. 5 – Type curve for $z = 0$ (uniform helix angle)



■ Fig. 6 – Type curve for $z = 1/4$

$$S = \frac{2\,M}{h} = 2\,G\,\frac{b\,t^3}{n\,R_l^3}\,Q$$

$$\tau_s = G\,t\,\frac{\alpha_s}{q\,R_l} \qquad \left(\frac{\eta_3}{\eta_2} \text{ approximated } = 1\right)$$

where:

b = blade width

t = blade thickness

n = number of free coils

R_l = coil radius at outer heel

α_s = helix angle at inner heel

q and z: as previously defined

F and Q are factors depending on q and z:

$$F = \left[z\,\frac{1-q^3}{3} + (1-z)\,\frac{1-q^2}{2}\right]\frac{2\pi}{1-q-(1-q)^2\,z}$$

$$Q = \frac{1}{\pi^2} \times \frac{1-q}{1-q^4} \qquad \left(\eta_2 \text{ approximated } = \frac{1}{\pi}\right)$$

These factors have been calculated in advance for the ranges of q and z. They are plotted in Figs. 10 and 11.

The present method neglects all except the main features; corrections may be subsequently introduced as refinements if desired. Among the neglected features are:

Stress correction for direct shear and curvature.

Secondary stresses.

Flexibility of end coils (usually allowed for by taking G somewhat too small).

Load eccentricity.

Accurate values for de Saint-Venant's coefficients η_3 and η_2 (changes in η_3 and η_2 will not change the type curves, only Q and τ).

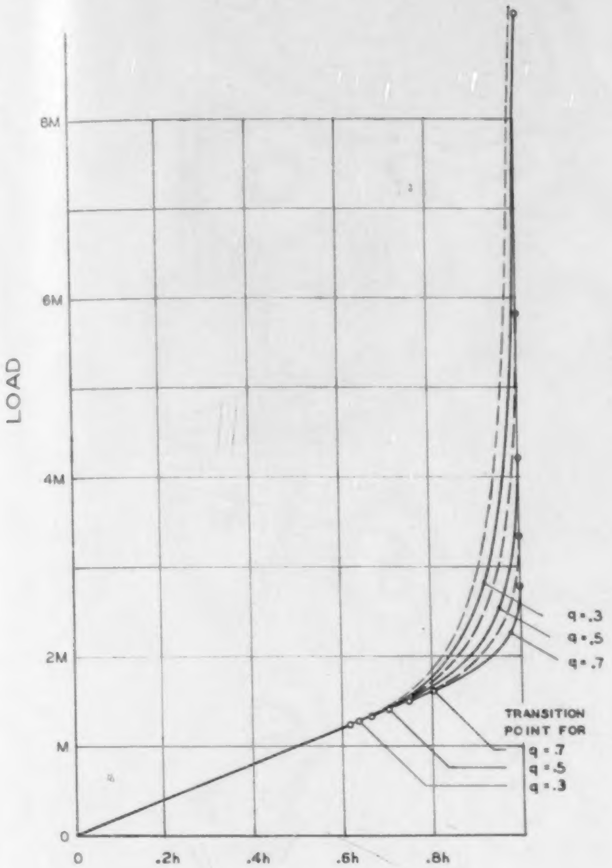
Tapering of blade in free coils.

Stresses – Layout of Finished Spring, Spring As Coiled, and Setting Bowl

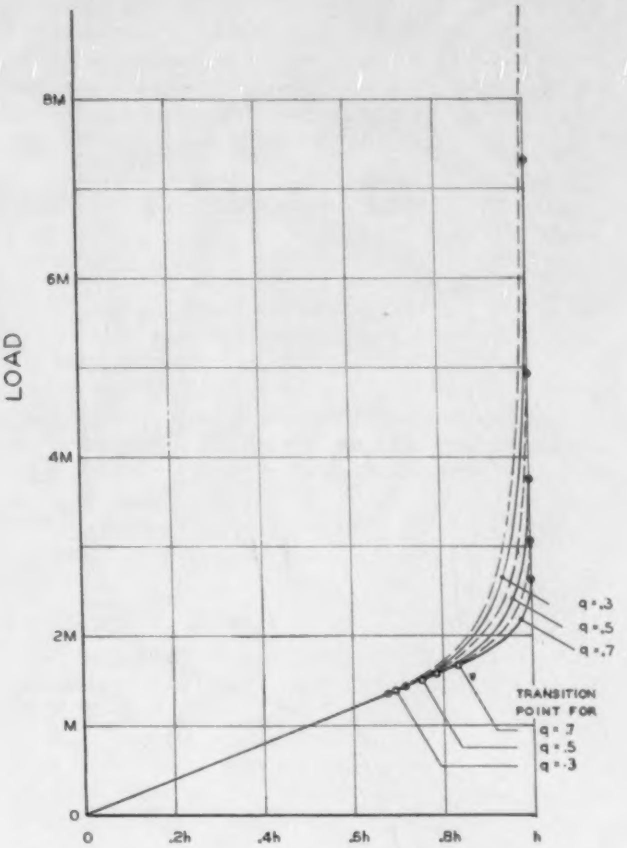
Most good springs show calculated and measured stress ranges beyond the yield stress of the material. Their ability to operate at these high stress ranges without failure depends entirely on residual or trapped stresses,[6] and consequently on the practice of giving the springs a permanent set as part of the manufacturing process.

As a result of the permanent set, the free spring is not free from stresses. The trapped or residual stresses are positive near the center of the cross-section, negative near the skin, and the resultant of their moments and shear forces is zero for any cross-section. The operating stress range can exceed the yield stress by the amount of the negative residual stress. If the spring has been preset to the greatest deflection which will later be imposed on it in service, the net stress (range less residual) will be the yield

[6] See "Strength of Materials," Part II, by S. Timoshenko, pp. 362-395: Chapter VIII, "Deformations beyond Elastic Limit," 1941, Van Nostrand, New York.



■ Fig. 7 – Type curve for $z = \frac{1}{2}$



■ Fig. 8 – Type curve for $z = \frac{3}{4}$

stress. If it has been preset to a greater deflection than it will undergo in service, the net service stress will be below the yield stress. (The yield stress itself may have been raised by strain hardening.)

The service life will be greatly affected by the residual stress: A stress range of 160,000 psi, with a net maximum of 110,000 psi, will produce failure much faster than the same stress range with a net maximum of 90,000 psi. The difference in life seems to be more pronounced in the region of limited life, which is of interest in suspension springs and bumpers, and less in the region of many millions of cycles, which is of interest for valve springs. Knowledge of Goodman diagrams for limited life or of equivalent data is a necessity if we want to use materials to the best advantage.[7]

With respect to residual stresses, the volute spring has two distinct advantages over the round-wire coil spring: It can be preset beyond solid height by setting it down into a bowl-shaped fixture such as is shown in Fig. 12, and the equilibrium of residual stresses in the rectangular cross-section of the free spring is such that a higher negative stress can be trapped near the surface than is possible with a round or square cross-section, because a smaller fraction of the area is subject to high strains. This explains the high load stresses which can be used with volute springs. For suspension springs, a maximum stress of 150,000 psi (without curvature correction) seems permissible, and for bumper springs, up to 190,000 psi.

It follows, from the importance of the residual stresses, that presetting is a manufacturing operation which deserves fully as much attention as the heat-treatment or the chemistry of the material.

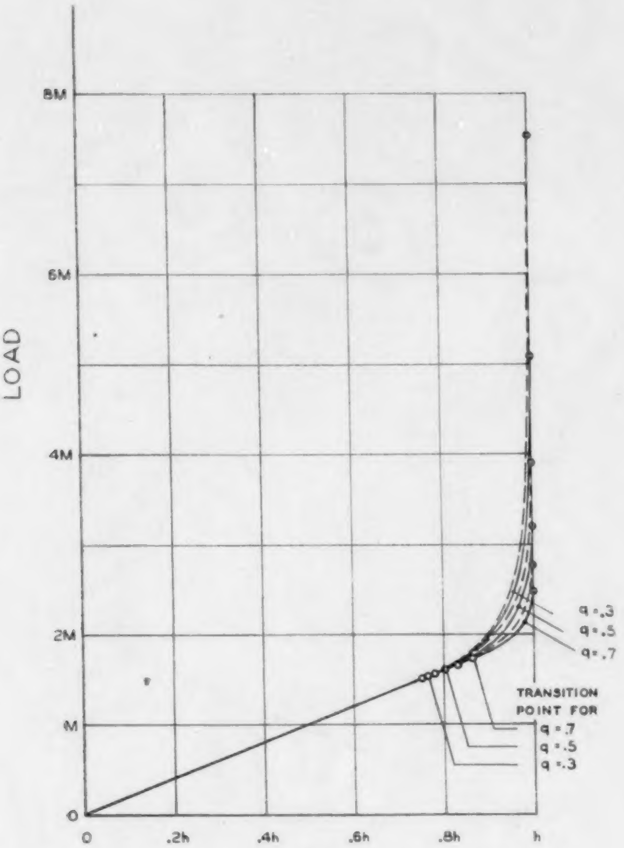
Two sets of springs which look exactly alike in every respect, are made of the same material, and had the same heat-treatment, may show entirely different test life, because one set may have been made by winding to some excess free height and presetting it onto a flat plate, and the other one may have been made by winding it to more excess free height and presetting into a bowl. Of the two, the spring which was preset into a bowl will be better.

A volute spring is, therefore, not defined unless the shape to which it was originally coiled and the shape of the bowl used in presetting are known, as well as the shape of the finished spring, the material, and its heat-treatment.

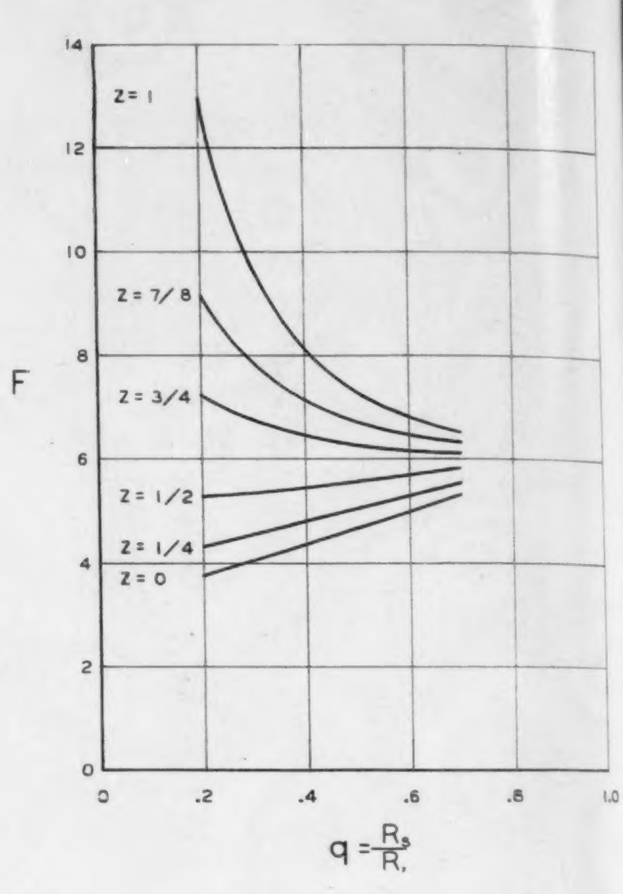
The study of presetting has only begun and the optimum practice cannot be prescribed yet, but the available experimental results seem to permit a few conclusions. These conclusions are here presented as a first attempt, which may and probably will be subject to revision as more data become available; meanwhile they may serve as a temporarily useful scaffolding to help in the creation of a more secure structure.

Presetting does no good unless it produces a permanent set. The permanent set at any point of a volute spring

[7] See SAE Transactions, Vol. 50, February, 1942, pp. 52-61: "Facts and Fallacies of Stress Determination," by J. O. Almen.

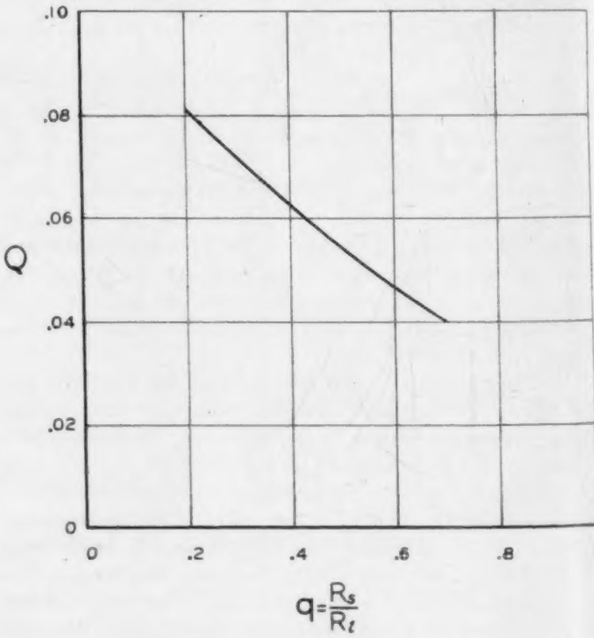


■ Fig. 9 – Type curve for $z = 1$ (uniform bottoming stress)



■ Fig. 10 – Auxiliary function F

$$\left(F = \left[\frac{z\,(1-q^3)}{3} + \frac{(1-z)\,(1-q^2)}{2}\right]\frac{2\pi}{1-q-(1-q)^2 z}\right)$$

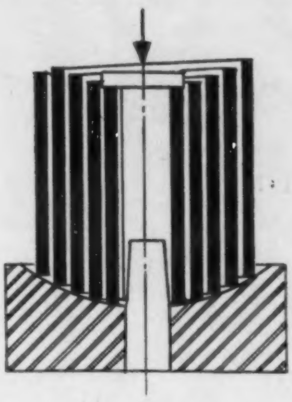


■ Fig. 11 – Auxiliary function Q $\left(Q = \frac{1-q}{\pi^2\,(1-q^4)}\right)$

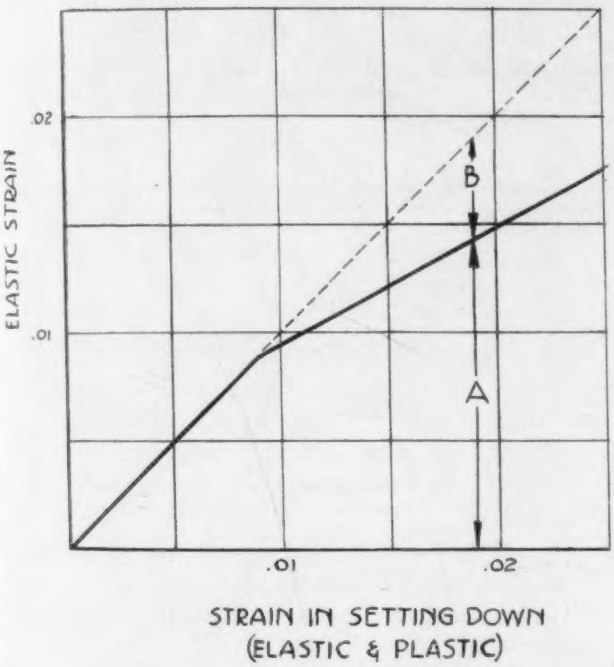
hot-coiled of silico-manganese steel to current practice has experimentally been found to be about 45% of the difference between presetting strain and original yield strain. The strain is calculated as $\beta t/R$ radians, where β is the change of helix angle from the free position to the fully compressed position in the bowl. The original yield strain is 0.009 radians, corresponding to a yield stress of 11,000,000 $\times$ 0.009 = 100,000 psi. A considerable amount of strain hardening is indicated by the proportionality between set and excess strain.

Fig. 13 shows this relation of permanent set to strain in setting down. The data were obtained by calculating strain in presetting and in recovery from helix angles measured at four sections on each coil of eight volute springs with blade thickness t about ⅜ in. Since the strains vary from section to section, the diagram is based on about eight-dozen test points which scatter around the given lines. This curve will not apply to different materials nor to the same material with different hardness and may not even apply to blades of much different thickness.

Such data can be used to design coiling form and presetting bowl of a volute spring. Assume for instance that a spring with $t = 0.3$ in., $R_l = 3$ in., $R_s = 1.5$ in., $\alpha_l = 0.16$ (radians), and $\alpha_s = 0.08$ (radians) shall be made, and that at every section the setting-down operation shall produce 0.002 radians more permanent set than setting the spring down flat would give.



■ Fig. 12 – Setting bowl



■ Fig. 13 – Permanent set of volute springs: A = elastic strain; B = permanent set; $A + B$ = total strain (blades about 3/8 in. thick, 430-475 Brinell)

The helix angles of the finished spring are called α.
The helix angles of the spring as coiled γ.
The helix angles of the spring pushed into the bowl θ.
The permanent set at any point will be

$$0.45\,[(\gamma + \theta)\,t/R - 0.009]$$

(from the empirical relation plotted in Fig. 13), and should be

$$0.45\,[\gamma t/R - 0.009] + 0.002.$$

(The expression in brackets of course must not be negative.)
Equating the two expressions and dividing by 0.45 gives:

$$(\gamma + \theta)\,t/R = \gamma\,t/R + 0.0044$$
$$\theta = 0.0044\,R/t = 0.0147\,R$$

This defines the shape of the setting bowl; its cross-section will be a cubic parabola, which can be approximated by an arc of a circle.

To find γ, the empiric relation:

$$(\alpha + \theta)\,t/R = (\gamma + \theta)\,t/R - 0.45\,[(\gamma + \theta)\,t/R - 0.009]$$
$$= 0.55\,(\gamma + \theta)\,t/R + 0.004$$

is used, and with $\theta = 0.0044\,R/t$ and $t = 0.3$, it yields

$$0.3\,\alpha/R + 0.0044 = 0.165\,\gamma/R + 0.0024 + 0.004$$
$$1.82\,\alpha/R - 0.012 = \gamma/R$$
$$\gamma = 1.82\,\alpha - 0.012\,R$$

The coiling mandrel or fixture should, therefore, be made with $\gamma_l = 1.82 \times 0.16 - 0.036 = 0.255$ radians at the large end, and $\gamma_s = 1.82 \times 0.16 - 0.018 = 0.127$ radians at the small end.

It is interesting to note that in this example both the finished spring and the spring as coiled give $z = 1$, that is, helix angles are proportional to coil radii. If the finished spring had been specified with $\alpha_s = \alpha_l = 0.10$, the spring as coiled would have to show:

$$\gamma_l = 0.182 - 0.036 = 0.146$$
$$\gamma_s = 0.182 - 0.018 = 0.164$$
$$z = \frac{1-1.12}{1-0.5} = -0.24 \text{ for the spring as coiled.}$$

(This spring would be stressed too highly; it is here given merely as an academic example to show how $z = 0$ can be produced.)

Defining the shape of springs and bowls in terms of helix angles is most convenient for the designer's preliminary calculations. To draw up the spring and to make the bowl and coiling guide, it is necessary to know the heights at various radii.

If for any radius R, the ratio $x = R/R_l$ is formed, and the height at that radius be called h_x, the formula:

$$h_x = n\,\alpha_l\,R_l\,\frac{2\pi}{1-q}\left[\frac{1-z}{2}\,(1-x^2) + \frac{z}{3}\,(1-x^3)\right]$$

gives the desired relation between radius and height. α_l can, of course, be replaced by γ_l or θ_l for mandrel and bowl respectively.

In practice, the original yield stress cannot be held constant. One heat of steel will not quench exactly as hard as another. To allow for these variations, a series of two or three bowls may be necessary, a shallower bowl for slightly softer springs and a deeper bowl for slightly harder springs.

■ Suggested Procedure for Use of Method

A. Analysis of a Given Spring

Measure or note:

coil radius at outer heel	R_l
coil radius at inner heel	R_s
blade width	b
blade thickness	t
number of free coils	n
total deflection	h
tangent of free helix angle at outer heel	α_l
tangent of free helix angle at inner heel	α_s
torsional modulus of elasticity	G

Calculate:

$$q = R_s/R_l$$
$$z = \frac{1 - \alpha_s/\alpha_l}{1 - q}$$

Find from Fig. 10: F (a function of q and z)
Find from Fig. 11: Q (a function of q)
Calculate:

$$h = n\,R_l\alpha_s F$$

which gives the deflection scale for the load-deflection curve. h must check with the measured total spring deflection. If necessary, the true distribution of α must be measured and approximated by a straight line.

$$M = G\,\frac{bt^3}{R_l^2}\,\alpha_s QF$$

which gives the load scale for the load-deflection curve.

$$S = \frac{2M}{h}$$

initial rate (straight-line part of the load-deflection curve).

Find from Figs. 5 to 9: the appropriate type curve defined by the calculated values of q and z. Interpolate if necessary.

Construct: the actual load-deflection curve by multiplying each abscissa value in the type curve with h and each corresponding ordinate value with M.

Calculate bottoming stresses:

at inner heel:

$$\tau_s = Gt \frac{\alpha_s}{R_s}$$

at outer heel:

$$\tau_l = Gt \frac{\alpha_l}{R_l}$$

at some intermediate point with coil radius $R = xR_l$:

$$\tau_x = Gt \frac{\alpha}{xR_l}$$

$$= \frac{1 - (1 - x)z}{x} \tau_l$$

Calculate loads (if desired):

transitional load (bottoming at outer heel):

$$P_l = \tau_l \frac{bt^2}{\pi R_l} \left(\text{assuming } \eta_2 = \frac{1}{\pi}\right)$$

Note: This load and the corresponding deflection are marked by a circle on the type curve.

load to bottom some intermediate point with coil radius $R = xR_l$:

$$P_x = \frac{P_l}{x} \frac{\tau_x}{\tau_l}$$

Note: For a really thorough check, it will be necessary to measure the helix angles along the entire blade. This will reveal to what extent the present method was in error in assuming a linear change of helix angle from coil to coil. The actual helix angles (which are likely to furnish variable z values throughout the spring) may be compared with the values assumed in the present method by plotting them into the diagram, Fig. 4.

Sample Analysis of a Given Spring

Measured and noted:

$R_l = 3.75$ $\quad b = 7.50$ $\quad n = 4$ $\quad G = 11 \times 10^6$
$R_s = 1.93$ $\quad t = 0.40$ $\quad h = 5.05$

α was measured (by height measurements) at every half coil. The results were plotted over R, and a straight line was drawn through the points (see Fig. 14). From this line was obtained:

$\alpha_l = 0.076$ $\qquad \alpha_s = 0.060$

Calculated:

$$q = \frac{1.93}{3.75} = 0.515 \qquad z = \frac{1 - 0.789}{1 - 0.515} = 0.434$$

Obtained from Fig. 10: $F = 5.5$

Obtained from Fig. 11: $Q = 0.053$

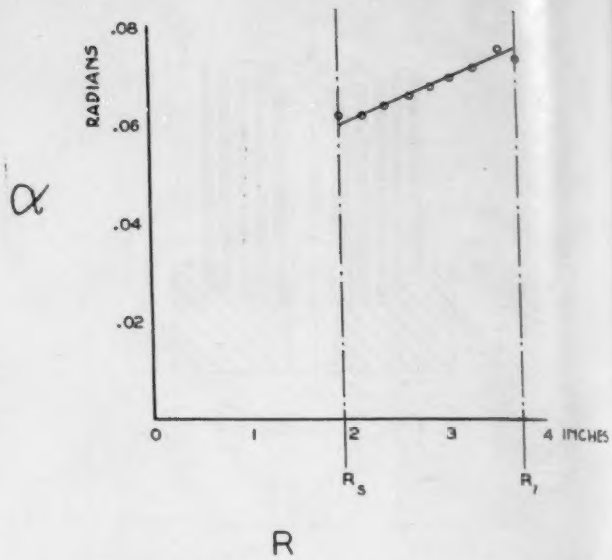
Calculated:

$h = 4 \times 3.75 \times 0.060 \times 5.5 = 4.95$ in. (This is satisfactory since it checks with the measured value within 2%.)

$$M = 11 \times 10^6 \times \frac{7.50 \times 0.064}{14.1} \times 0.060 \times 0.053 \times 5.5$$

$= 6550$ lb. (This is satisfactory since the measured load at 2.5 in. deflection was 6500 lb.)

With h and M, the scales for the type curve are given. The correct type curve would be between two curves plotted for $z = 0.25$, $q = 0.5$; and for $z = 0.50$, $q = 0.5$.



■ **Fig. 14 – Measured distribution of helix angles in a volute spring, plotted against coil radius**

The difference between these two curves amounts nowhere to more than a shift by 0.1 in. deflection, which is less than the hysteresis loop.

Calculated:

$$S = \frac{6550}{2.5} = 2620 \text{ lb per in.}$$

$$\tau_s = 11 \times 10^6 \times 0.40 \times \frac{0.060}{1.93} = 137{,}000 \text{ psi}$$

$$\tau_l = 11 \times 10^6 \times 0.40 \times \frac{0.076}{3.75} = 89{,}000 \text{ psi}$$

B. Design of a New Spring

Choose:

q and z. This involves a compromise between rate build-up and distribution of bottoming stresses.

The bottoming stresses are more nearly uniform in all coils for higher values of z and q.

However, at high values of z, the distribution of bottoming stresses is not seriously impaired by lower values of q. On the other hand, at low values of z, the distribution of bottoming stresses becomes rapidly more uneven with decreasing q, so much so that the addition of steel in the form of inner coils may actually raise the maximum stress instead of lowering it.

High rate build-up at the end of the stroke can be obtained at the cost of uneven distribution of bottoming stresses by using low values of q and z.

Choose:

τ_s maximum torsional stress. (Depending on material and use.)

Note: G torsional modulus of elasticity.

(For hot-coiled steel blades, a value of 11,000,000 psi is generally used.)

Choose or determine:

R_l coil radius at outer heel.

(Smaller than ½ O.D. to allow for outer end coil.)

h necessary total deflection.

M load at ½ of total deflection.

(This may be determined by choosing the initial rate S, which equals $2M/h$, and which may also be considered as static load divided by static deflection.)

Find from Fig. 10: F (a function of q and z).

Find from Fig. 11: Q (a function of q).

Calculate:

$$bt^2 = \frac{MR_l}{QFq\tau_s} \quad \text{(from } M \text{ and } \tau_s \text{ formulas).}$$

$$\alpha_s t = \frac{qR_l\tau_s}{G} \quad \text{(from } \tau_s \text{ formula).}$$

$$n\alpha_s = \frac{h}{FR_l} \quad \text{(from } h \text{ formula).}$$

These three equations furnish the interrelation between the unknown values of b, t, α_s and n.

The first equation serves to choose t and determine b, or vice versa.

Then α_s can be determined from the second equation, and finally n from the third equation.

In order to arrive at satisfactory proportions, it will generally be necessary to try several combinations of b and t from which to select one combination.

The radial pitch is $\dfrac{R_l - R_s}{n}$.

This must leave a satisfactory gap between coils.

Construct: The load-deflection curve in the manner outlined under previous section *"A. Analysis of a Given Spring."*

Calculate: Stresses and loads. (Same as outlined under previous section *"A. Analysis of a Given Spring."*)

Sample Design of a New Spring

A bumper spring is desired with:

S = 1000 lb per in.	considerable rate build-up
h = 2.80 in.	maximum O.D. = 3⅞ in.
τ_s = 180,000 psi	minimum I.D. = 1 in.

A trial estimate is made with:

R_l = 1.70 in. and R_s = 0.60 in., giving q = 0.353, and with a choice of z = ½ to obtain the desired rate build-up.

$$M = \frac{1000 \times 2.80}{2} = 1400 \text{ lb},$$

giving the load scale on the load-deflection curve.

Obtained from Fig. 10: F = 5.4

Obtained from Fig. 11: Q = 0.067

$$\text{Calculated: } bt^2 = \frac{1400 \times 1.70}{0.067 \times 5.4 \times 0.353 \times 180{,}000} = 0.1035$$

$$\alpha_s t = \frac{0.353 \times 1.70 \times 180{,}000}{11{,}000{,}000} = 0.00985$$

$$n\alpha_s = \frac{2.80}{5.4 \times 1.70} = 0.305$$

Trial combinations:

t	0.125(⅛)	0.156(5/32)	0.188(3/16)
t^2	0.0156	0.0244	0.0354
$b = 0.1035/t^2$	6.64	4.24	2.92
$\alpha_s = 0.00985/t$	0.0788	0.0632	0.0524
$n = 0.305/\alpha_s$	3.88	4.83	5.82
pitch = $(1.70 - 0.60)/n$	0.284	0.228	0.189

The trial calculations show that the gap between coils becomes too small for springs with t = 3/16 or larger. With t = 5/32, the gap is ample and the blade is not too wide. If desired, n may be modified for such considerations as material saving, load eccentricity, and available tools.

Here n may be reduced to 4½, the only consequence being that α_s will be increased to 0.0679 and τ_s to 194,000. If this stress is considered too high, R_s may be increased to 0.65 (with t remaining at 5/32). Then

q = 0.382

F = 5.45 (from Fig. 10)

Q = 0.064 (from Fig. 11)

$$bt^2 = \frac{1400 \times 1.70}{0.064 \times 5.45 \times 0.382 \times 180{,}000} = 0.0995$$

$$\alpha_s t = \frac{0.65 \times 180{,}000}{11{,}000{,}000} = 0.01065$$

$$n\alpha_s = \frac{2.80}{5.45 \times 1.70} = 0.302$$

$$b = \frac{0.0995}{0.0244} = 4.07 \qquad \alpha_s = \frac{0.01065}{0.156} = 0.0685$$

$$n = \frac{0.302}{0.0685} = 4.42 \qquad \text{pitch} = \frac{1.70 - 0.65}{4.42} = 0.238$$

The inside diameter, with the usual 3/4 tapered inner end coil, will be

I.D. = $2R_s$ − pitch = 1.30 − 0.24 = 1.06, which is satisfactory.

The outside diameter, with one full turn of full thickness outer end coil, will be

O.D. = $2R_l + t$ + 1½ pitch = 3.40 + 0.16 + 0.36 = 3.92, which is slightly above 3⅞, but which can be reduced, in manufacture, by 0.050 in.

Appendix I

Symbols

Subscript l refers to outer heel (large end).

Subscript s refers to inner heel (small end).

Subscript x refers to the section with coil radius $R = xR_l$.

R = coil radius

α = (tangent of) free helix angle

n = number of free coils

h = free height above outer heel

b = blade width

t = blade thickness

L = blade length

P = load

P_x = load to bottom section at $R = xR_l$

f = deflection

f_x = deflection under load P_x

τ = torsional stress

τ_x = bottoming stress at $R = xR_l$

τ_{Px} = stress at R under load $P < P_x$

G = torsional modulus of elasticity (taken as 11,000,000 psi for hot-coiled steel)

η_2 = de Saint-Venant's stress coefficient
η_3 = de Saint-Venant's deflection coefficient
} by approximation $\eta_2 = \eta_3 = \dfrac{1}{\pi}$

q = radius ratio = R_s/R_l

z = relative rate of change of helix angle

$$= \frac{1 - \alpha_s/\alpha_l}{1 - R_s/R_l}$$

$$= \text{(by assumption) } \frac{1 - \alpha/\alpha_l}{1 - R/R_l}$$

$$Q = \frac{1}{\pi^2} \times \frac{1 - q}{1 - q^4} \quad \text{(a useful factor, plotted out on Fig. 11)}$$

$$F = \frac{2\pi}{1 - q - (1 - q)^2 z}\left[z\,\frac{1 - q^3}{3} + (1 - z)\,\frac{1 - q^2}{2}\right]$$

(a useful factor, plotted out on Fig. 10)

Appendix II

Table of Formulas

For volute springs with constant or uniformly varying helix angle, and with uniform radial pitch

		General Formulas	Simplified Formulas for use with given diagrams
HELIX ANGLES			
free helix angle — at outer heel	α_l	$\frac{1}{1-z+qz}\,\alpha_s$	
free helix angle — at inner heel	α_s	$(1-z+qz)\,\alpha_l$	
free helix angle — at section with coil radius $R = xR_l$	α	$(1-z+xz)\,\alpha_l$	
STRESSES			
bottoming stress — at outer heel	τ_l	$G\,\frac{\eta_3}{\eta_2}\,t\,\frac{\alpha_l}{R_l}$	$Gt\,\frac{\alpha_l}{R_l}$
bottoming stress — at inner heel	τ_s	$G\,\frac{\eta_3}{\eta_2}\,t\,\frac{\alpha_s}{R_s} = \left(z + \frac{1-z}{q}\right)\tau_l$	$Gt\,\frac{\alpha_s}{R_s}$ $\quad \frac{\eta_3}{\eta_2} \approx 1$
bottoming stress — at section with coil radius $R = xR_l$	τ_x	$G\,\frac{\eta_3}{\eta_2}\,t\,\frac{\alpha}{R} = \left(z + \frac{1-z}{x}\right)\tau_l$	$Gt\,\frac{\alpha}{R}$ $\quad \eta_3 \approx \frac{1}{\pi}$
load stress without bottoming	τ_{P_x}	$P\,\frac{R}{\eta_2 bt^2}$	
LOADS			
load at deflection ½ h	M	$Gbt^3\,\frac{\alpha_s}{R_l^2}\,\frac{\eta_2}{1-q^4}\,\frac{2}{1-z+qz}\left[z\,\frac{1-q^3}{3} + (1-z)\,\frac{1-q^2}{2}\right]$	$Gbt^3\frac{\alpha_s}{R_l^2}\,QF$
transition load	P_l	$G\eta_3 bt^3\,\frac{\alpha_l}{R_l^2}$	circled point on type curve
load to bottom section at radius $R = xR_l$	P_x	$G\eta_3 bt^3\,\frac{\alpha}{R^2} = \frac{1-z+xz}{x^2}\,P_l$	
final bottoming load	P_s	$G\eta_3 bt^3\,\frac{\alpha_s}{R_s^2} = \frac{1-z+qz}{q^2}\,P_l$	circled point on type curve
DEFLECTIONS			
total deflection	h	$nR_l\alpha_l\,\frac{2\pi}{1-q}\left[z\,\frac{1-q^3}{3} + (1-z)\,\frac{1-q^2}{2}\right]$	$nR_l\alpha_s F$
transition deflection	f_l	$nR_l\alpha_l\,\frac{\pi}{2}\,\frac{1-q^4}{1-q}$	$\frac{nR_l\alpha_l}{2\pi Q}$ circled point on type curve
deflection to bottom section at radius $R = xR_l$	f_x	$nR_l\alpha_l\,\frac{2\pi}{1-q}\left[(1-z+xz)\,\frac{x^4-q^4}{4x^2} + z\,\frac{1-x^3}{3} + (1-z)\,\frac{1-x^2}{2}\right]$	
free height of section at radius $R = xR_l$	h_x	$nR_l\alpha_l\,\frac{2\pi}{1-q}\left[z\,\frac{1-x^3}{3} + (1-z)\,\frac{1-x^2}{2}\right]$	
RATE			
initial rate before any bottoming	S	$\frac{2\eta_3}{\pi}\,\frac{Gbt^3}{nR_l^3}\,\frac{1-q}{1-q^4}$	$2G\,\frac{bt^3}{nR_l^3}\,Q$

Appendix III

Modifications for Unusual Sections

The type curves and the diagram for F apply equally well to all cross-sections.

The factor Q depends on the assumption $\eta_3 = 1/\pi$. Where this assumption does not apply, particularly for other than narrow rectangular blades ($b > 5t$) a correction can be made by multiplication with $\pi\eta_3$.

For square wire this correction factor is 0.440.

For round wire this correction factor is 0.308.

Appendix IV

Derivation of Formulas

With the assumption of linear change of helix angle with coil radius (as shown under section "Main Features of the Volute").

$$\frac{1-\alpha/\alpha_l}{1-x} = z$$

$$\alpha_l - \alpha = (z - xz)\,\alpha_l$$

$$\alpha = (1 - z + xz)\,\alpha_l \qquad (1)$$

$$\alpha_s = (1 - z + qz)\,\alpha_l \qquad (1A)$$

The bottoming stress at any point (without correction for curvature, end coil flexibility, deviation of spiral arc from circular arc, secondary stresses, eccentricity, and so forth) is

$$\tau_x = G \frac{\eta_3}{\eta_2} t \frac{\alpha}{R} \quad \text{(from windup)}$$

The ratio of bottoming stress at inner heel to bottoming stress at outer heel is then

$$\tau_s/\tau_l = \frac{\alpha_s R_l}{\alpha_l R_s} = \frac{1 - z + qz}{1} \times \frac{1}{q} = z + \frac{1 - z}{q} \quad (2)$$

The efficiency of the spring (equal service life expectancy at all sections) depends on this ratio.

For $z = 1$, the ratio is $\tau_s/\tau_l = 1$. The bottoming stress is then uniform, regardless of radius ratio q.

For $z = o$ (uniform helix angle) the ratio is $\tau_s/\tau_l = 1/q$. This can become very unfavorable for small values of q.

At an intermediate point, the bottoming stress will be

$$\tau_x = \frac{(1 - z + xz)}{x} \tau_l = \left(z + \frac{1 - z}{x}\right) \tau_l \quad (2A)$$

The stress can also be calculated from torque and section modulus as

$$\tau_{Px} = P \frac{R}{\eta_2 bt^2}$$

Thus, before bottoming, the stresses in the spring decrease from outer heel to inner heel in proportion to the coil radius. After bottoming, the stresses usually increase from outer heel to inner heel (in accordance with equation 2A).

When a section is just bottoming under its bottoming load P_x, the two ways of calculating the stress must yield the same result. This equality gives an equation for P_x.

$$\tau_x = \frac{P_x R}{\eta_2 bt^2} = G \frac{\eta_3}{\eta_2} t \frac{\alpha}{R}$$

$$P_x = G\eta_3 bt^3 \frac{\alpha}{R^2} \quad (3)$$

With $R = xR_l$ and the assumption $\alpha = (1 - z + xz)\alpha_l$, this becomes

$$P_x = G\eta_3 bt^3 \frac{\alpha_l}{R_l^2} \frac{(1 - z + xz)}{x^2} \quad (3A)$$

The transition load P_l, where bottoming just begins, is

$$P_l = G\eta_3 bt^3 \frac{\alpha_l}{R_l^2} \quad (3B)$$

Therefore

$$P_x = \frac{1 - z + xz}{x^2} P_l \quad (3C)$$

With $z = 0$, the bottoming loads increase in inverse proportion to the square of the coil radii.

With $z = 1$, the bottoming loads increase in inverse proportion to the coil radii.

Up to the load P_l, the deflections f are proportional to the loads.

After bottoming has started, the deflections f_x are composed of two parts:

$$f_x = f_x' + f_x''$$

f_x' is the deflection of the bottomed coils, equal to the free height h_x of the section at $R = xR_l$.

f_x'' is the deflection of the coils which are not bottomed.

$$f_x' = h_x = \int_0^{L_x} \alpha_r dL$$

α_r is the helix angle at a point with radius $R_r = rR_l$. It is necessary to distinguish

between x, which is the index of the last bottomed section,

and r, which is the index of any section and the variable of integration.

$\alpha_r = (1 - z + rz)\alpha_l$ (from equation 1).

The radius at the near end of dL is rR_l.

The radius at the far end of dL is $(r - dr)R_l$.

The radius changes from R_l to R_s ($= qR_l$) in n turns.

The arc corresponding to dr is, therefore,

$$\frac{2\pi n}{1 - q} dr \quad \text{and} \quad dL = rR_l \frac{2\pi n}{(1 - q)} dr.$$

The limits of the integral must be expressed in terms of r instead of L.

At the outer heel $L = 0$ and $r = 1$.

At the section x, $L = L_x$ and $r = x$.

Since r decreases as L increases, the negative of the integral must now be taken. This is done by interchanging the upper and lower limit of integration.

Thus

$$f_x' = h_x = \frac{2\pi n}{1-q} R_l \alpha_l \int_x^1 (1 - z + rz)\, r\, dr$$

$$f_x' = h_x = \frac{2\pi n}{1-q} R_l \alpha_l \left[(1 - z)\frac{(1 - x^2)}{2} + z\frac{(1 - x^3)}{3}\right] \quad (4)$$

For any coil element which is not yet bottomed, the helix angle will have decreased by an amount β_r, which is the same fraction of the free helix angle α_r as P_x is of P_r.

$$\beta_r = \frac{P_x}{P_r} \alpha_r$$

$$\frac{P_x}{P_r} = \frac{(1 - z + xz)}{x^2} \times \frac{r^2}{(1 - z + rz)} \quad \text{(from equation 3C)}$$

$$\alpha_r = (1 - z + rz)\alpha_l \quad \text{(from equation 1)}$$

$$\beta_r = \frac{(1 - z + xz)}{x^2} r^2 \alpha_l$$

$$f_x'' = \int_{L_x}^{L_q} \beta_r dL$$

$$f_x'' = \frac{2\pi n}{1 - q} R_l \alpha_l \frac{(1 - z + xz)}{x^2} \int_q^x r^3 dr$$

$$f_x'' = \frac{2\pi n}{1 - q} R_l \alpha_l (1 - z + xz) \frac{x^4 - q^4}{4x^2}$$

Adding the result for f_x'' to f_x' as given by equation 4 yields

$$f_x = nR_l \alpha_l \frac{2\pi}{1 - q} \left[(1 - z + xz)\frac{x^4 - q^4}{4x^2} + z\frac{1 - x^3}{3} + (1 - z)\frac{1 - x^2}{2}\right] \quad (5)$$

The transition deflection at which bottoming just begins is obtained when $x = 1$. It is

$$f_l = nR_l \alpha_l \frac{\pi(1 - q^4)}{2(1 - q)} \quad (5A)$$

The total possible deflection is obtained when the inner heel is bottomed ($x = q$). It is

$$f_s = h = nR_l \alpha_l \frac{2\pi}{(1-q)} \left[z\frac{1 - q^3}{3} + (1 - z)\frac{1 - q^2}{2}\right] \quad (5B)$$

For convenience in application, where α_s is usually the limiting factor because of stress, α_s is substituted from equation 1A. Then

$$h = nR_l \alpha_s \frac{2\pi}{1 - q - (1 - q)^2 z} \left[z\frac{1 - q^3}{3} + (1 - z)\frac{1 - q^2}{2}\right] \quad (5C)$$

The expression

$$F = \frac{2\pi}{1 - q - (1 - q)^2 z} \left[z\frac{1 - q^3}{3} + (1 - z)\frac{1 - q^2}{2}\right]$$

has been calculated and plotted on Fig. 10. Therefore, with the use of this plotted function:

$$h = nR_l \alpha_s F \quad (6)$$

Up to the transition load P_l, and transition deflection f_l, the load-deflection curve is a straight line. The slope of this line is the initial rate S. From equations 3B and 5A

$$S = \frac{P_l}{f_l} = \frac{2\eta_3 G b t^3 \alpha_l (1-q)/R_l^2}{n R_l \alpha_l (1-q^4)\pi}$$

$$S = 2G \frac{bt^3}{nR_l^3} \frac{\eta_3}{\pi} \frac{1-q}{1-q^4} \tag{7}$$

For the usual blade proportions, η_3 may be approximated by $1/\pi$.

The factor

$$Q = \frac{1}{\pi^2} \frac{(1-q)}{(1-q^4)}$$

has been calculated and plotted on Fig. 11. Therefore, with the use of this plotted function

$$S = 2G \frac{bt^3}{nR_l^3} Q \tag{7A}$$

In the type curves, all loads are expressed as multiples of the load at $f = \frac{1}{2}h$. This load is, from equations 6 and 7A:

$$M = \tfrac{1}{2}Sh = G \frac{bt^3}{R_l^2} \alpha_s QF \tag{8}$$

The type curves were constructed by calculating pairs of corresponding values for f_x/h and P_x/M. The values of G, b, t^3, R_l, α, n, η cancel. Then

$$\frac{fx}{h} = \frac{(1-z+xz)\dfrac{x^4-q^4}{4x^2} + z\dfrac{1-x^3}{3} + (1-z)\dfrac{1-x^2}{2}}{z\dfrac{1-q^3}{3} + (1-z)\dfrac{1-q^2}{2}}$$

$$\frac{P_x}{M} = \frac{P_x}{P_l} \times \frac{P_l}{M} = \frac{1-z+xz}{x^2} \times \frac{\alpha_l}{\alpha_s} \times \pi \times \frac{1-q^4}{1-q} \times \frac{1}{F}$$

$$\frac{P_x}{M} = \frac{1-z+xz}{1-z+qz} \times \frac{1}{x^2 QF\pi}$$

DISCUSSION

Emphasizes Need for Figures To Show Effect of Presetting

– D. J. LaBelle

Yellow Truck & Coach Mfg. Co.

MR. Fuchs presents a valuable contribution not only to the methods of design but to at least one step in the manufacture of volute springs, which tends to minimize its greatest drawback, namely, its limited life.

1. *Design Methods* – The idea of using nondimensional factors in the initial stages of design of any structural member represents an ideal which is the goal of any stress analyst. To the ordinary designing engineer, however, for whom the use of volute springs is now quite limited, time may actually be saved by going through the tedious but simple calculations and cut-and-try methods rather than applying nondimensional factors, which sometimes are quite vague and require a good deal of study.

2. *Manufacturing Methods* – Presetting volute springs into a bowl and the presentation of the theory involved is the outstanding phase of the report. "Trapping" of stresses in the core of the spring material puts this formerly idle metal to work, and moving the stress range on the outer skin down the scale will undoubtedly increase the spring life.

3. *General Comments* – Addition to the report of some figures showing the effect of presetting on the increase of spring life would not only be interesting but would substantiate the presetting theory.

It would also be interesting to learn if any progress is being made to maintain positive intercoil clearance to prevent friction between coils, with the accompanying stress concentrating abrasions which probably decrease spring life.

Navy Maintenance

A DIESEL engine requires competent operators if it is to function at designed efficiency and dependability. A few years ago, it was possible to do practically all our training on board ship, where new engineers could be gradually taught the whys and wherefores of an engine under the guidance of more experienced men. With expansion, this system was definitely too slow. High-pressure training was necessary.

We have a school to which are sent students with at least a faint knowledge of diesel engines. In the six weeks allotted we can do little except attempt to give them the details of the particular type of engine they can expect to encounter when they leave. We try to make them understand that if they will but keep their engines clean, properly lubricated, and adjusted, the machines will give excellent service and a large portion of our maintenance problems will be eliminated.

We do have a maintenance problem – of even greater magnitude than in peace. It is somewhat hard to realize the troubles that can arise when a repair job comes up many thousands of miles and perhaps months of shipping time away from manufacturers – and all repairs have to be made from material on hand or improvised.

■ Replacement Parts

Because of the delays in shipping and the possibilities of losses through warfare, we must anticipate by many months our wants of the parts normally required for an engine overhaul and routine repairs.

We of the Navy realize that our use of replacement parts is excessive. Some of that excess can be placed squarely on the shoulders of our all too little experienced personnel. Realizing their own lack of knowledge, rather than reinstall parts that show signs of wear, they discard bearings and the like when they have several thousand engine hours of useful service left in them. That lack of knowledge is something that we are making every effort to overcome by education and training.

We cannot afford to have engines break down – that is why such frequent inspections and overhauls are a routine and a necessity on our ships. Granted that at times we may be justly accused of wearing out our engines, just tearing them down for an inspection to determine if everything is still all right. Some of it is unnecessary, we all know, but at the same time we cannot afford to have an engine out of commission when a ship is at sea. To have such an engine fail due to lack of inspection or replacement of a doubtful part would be little short of criminal.

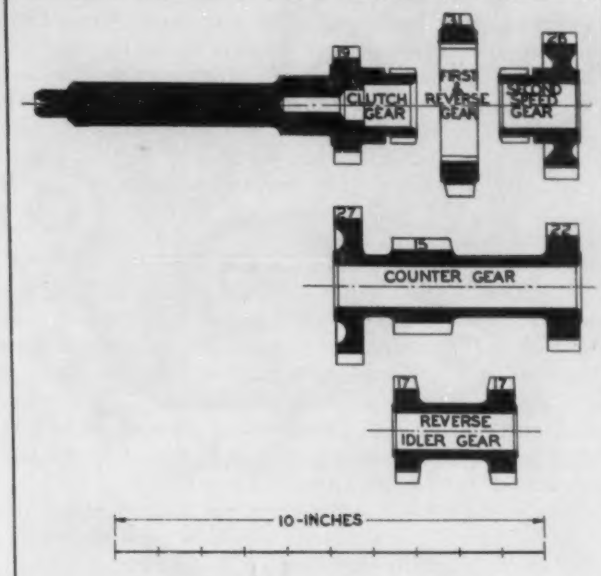
So let me confess for the Navy that we realize thoroughly we are far from perfect. But we are learning fast – and the hard way. Our maintenance crews are becoming more expert with experience, our overhaul facilities and machinery are improving rapidly – and if the engine manufacturers will but continue to give us the grand engines they are turning out today, we of the Navy will keep them running.

Excerpts from "Diesel Maintenance in the Fleet," by Lt.-Com. R. J. Moore, USN, Cleveland Diesel-Engine Division, General Motors Corp., presented at the SAE Diesel Engine and Fuels and Lubricants Meeting, Cleveland, Ohio, June 3, 1943.

The POSSIBILITIES of SHAVED GEARS for AIRCRAFT ENGINES

by A. W. HARRIS

Chevrolet Motor Division, General Motors Corp.



■ Fig. 1 – Typical transmission gears

ALTHOUGH shaving had already become a common method of finishing spur and helical gears in the automotive industry before the war, there were many problems that had to be solved before this method could be applied to aircraft-engine gears.

The major objection of the aircraft industry to shaving has been the probability of distortion during heat-treatment subsequent to finishing the tooth form. Mr. Harris suggests that the heat-treatment procedure be changed to make expansion more uniform, and then an allowance be introduced in cutting the gear originally to compensate for this uniform expansion.

Another objection to shaving has been the probability of highly concentrated stresses due to cutter marks and the line of demarcation between the hobbed and shaved contour in the tooth fillet. Mr. Harris proposes a form of hob for giving smooth fillet contours that will blend with the shaved active profile of the gear.

The author also discusses the types of errors that might occur in the gear before shaving and be only partially corrected.

ABOUT 12 years ago, Chevrolet incorporated the operation of shaving in their production procedure for finishing transmission gears. This shaving was done on both rack- and rotary-types of machines. The purpose was to improve the quality of the gears by getting a smoother tooth contact surface and to eliminate the matching of gears by decreasing the variation from an established standard form. The development work of the manufacturers of these two machines was so successful that within a few years shaving had become the common method of finishing spur and helical gears in the automotive industry.

Today, the War Department is asking us to use this method of finishing gears wherever practicable for the aircraft engine we build. This time, the change in procedure is for the purpose of relieving an anticipated equipment bottleneck by spreading the requirements over more types of machines.

The introduction of any method of manufacture as different as shaving is from grinding naturally brings up a multitude of incidental problems. This report of some of our experience with shaving transmission gears shows the type of problems we expect in shaving aircraft-engine gears.

■ Expansion of Gears During Hardening

The major objection to shaving in the aircraft industry seems to center around the probability of distortion during heat-treatment subsequent to finishing the gear tooth form. That word "distortion" is one I do not like. Considering it rather as expansion, sometimes uniform and sometimes non-uniform, a method of control suggests itself. Heat-treating procedure can be changed to make it more uniform and an allowance introduced to compensate for the uniform expansion in cutting the gear originally.

During the development of the 1937 Chevrolet trans-

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 9, 1943.]

THE AUTHOR: A. W. HARRIS (M '30) was first employed by the Cadillac Motor Co. from 1916 to 1918 as blueprint boy, tracer, tool designer and squad leader. Then followed a period of various positions with such companies as Studebaker, Nelson Blank Mfg. Co., Chevrolet's Toledo Division, and Chevrolet's Transmission Division at Saginaw, as resident engineer in charge of activities on transmissions at Saginaw, Muncie, and Toledo. In 1941 he joined the Chevrolet Aviation Engine Plant at Tonawanda, N. Y., as assistant chief engineer responsible for engineering contacts at manufacturing plants.

mission, we tried to hold down the hardness of the internal splined surface on the first and reverse gear to permit rebroaching after hardening. The gears were clamped on a splined arbor between copper washers during the hardening operations. After hardening, the expansion rate was found to be of the nature of +0.0050 on the diameters. The 2⅝-in. inside diameter expanded 0.013. Since the gear and spline expanded together, we gave up the idea of hard broaching and concentrated our efforts on controlling the expansion. This decision may also have been influenced by the fact that the internal spline hardness was 47 "C" Rockwell, in spite of our precautions.

Later, we made a series of checks on transmission gear sets to determine whether the volume change during heat-treatment was a uniform expansion. Production gears made of SAE 5145 steel were hardened by immersion in a cyanide bath at 1490 F for approximately 12 min and then quenched in oil. Development gears made of SAE 4815 steel were carburized at 1675 F, cooled in the box, reheated to 1470 F, and quenched in oil.

The involute form of the gears was charted before and after hardening. The change was then converted into a corresponding change of base circle diameter and compared with the change of other gear elements.

The gear sections are shown in Fig. 1, and the expansion rates in Table 1. While it will be noted that all elements of the SAE 5145 steel gears did not expand uniformly, it can be seen that the change in the tooth form is part of the general expansion of the gear during heat-treatment. On the basis of the experience cited previously, we believe that changes in the holding fixtures used during heating and quenching to control the cooling rate of the gear

Table 1 – Expansion Rates During Heat-Treatment of Passenger and Commercial Transmission Gears

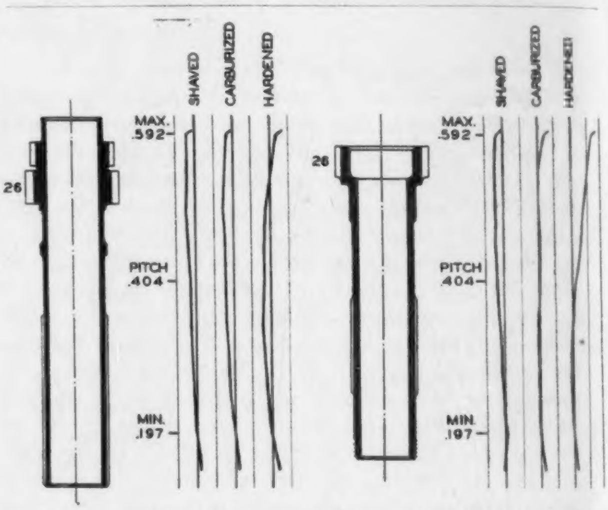
	Passenger SAE 5145 Steel	Commercial SAE 4815 Steel
Clutch Gear		
Base circle	+0.0010	−0.0011
Dimension over gage blocks	+0.0010	+0.0012
Dimension over rolls in clutch teeth	+0.0007	+0.0007
Bearing diameter	+0.0010	−0.0002
Bore	+0.0006	−0.0040
First and Reverse Gear		
Base circle	+0.0011	−0.0002
Dimension over rolls in teeth	+0.0010	+0.0011
Dimension between rolls in spline	+0.0016	+0.0001
Width	+0.0022	−0.0002
Second-Speed Gear		
Base circle	+0.0012	−0.0018
Dimension over gage blocks	+0.0014	+0.0011
Dimension over rolls in clutch teeth	+0.0011	+0.0006
Length	+0.0012	−0.0007
Outside diameter	+0.0012	−0.0007
Bore	+0.0009	−0.0033
Countergear		
27-tooth base circle	+0.0010	−0.0016
Dimension over gage blocks	+0.0011	+0.0006
15-tooth base circle	+0.0011	−0.0011
Dimension over rolls in teeth	+0.0010	+0.0007
22-tooth base circle	+0.0010	−0.0010
Dimension over gage blocks	+0.0010	+0.0010
Outside diameter	+0.0013	−0.0008
Length	+0.0010	−0.0010
Bore	+0.0005	−0.0031
Idler Gear		
Base circle	+0.0012	−0.0010
Dimension over rolls in teeth	+0.0011	+0.0006
Diameter between gears	+0.0009	−0.0011
Length	+0.0010	−0.0010
Bore	+0.0009	−0.0037

would have resulted in a uniform expansion. However, since the gear teeth themselves expanded quite uniformly, we felt that changes to control the other elements were not justified. On the basis of these results, confirmed by numerous spot checks at both plants manufacturing these gears, an allowance for expansion during heat-treatment of +0.0010" was established for the SAE 5145 steel gears.

The SAE 4815 steel gears did not expand as uniformly as the others. All elements except the measurements over rack-shaped gaging blocks between the gear teeth of helical gears, and over rolls for spur gears, showed a minus expansion rate. A tentative allowance for expansion during heat-treatment of −0.0010 was established for these gears. Our inability to understand the reasons for the variations, and the low production required, restricted efforts to control this expansion. It was never maintained as uniformly as on the SAE 5145 steel. One variation we were aware of, but had no means of controlling, was the depth of case, which was influenced by the location in the carburizing box.

The fact that aircraft-engine gears are normally carburized only locally and the case depth more closely controlled will probably make them expand more uniformly. We can give you expansion data on the tooth form of one gear we are now shaving and carburizing for aircraft-engine use.

The impeller shaft gear shown in Fig. 2 has 26 teeth of 11/13 diametral pitch. The vertical points on the tooth chart indicated are the lowest point of contact with the mate, the pitch line, and the tip of the tooth. The dimensions are distances along the line of action from the point of tangency with the base circle. The spaces between adjacent vertical lines represent 0.0005 each. The gear on the left is being shaved in production on a rotary shaver, carburized in a gas carburizer, and hardened in an electric furnace. The bulge near the pitch line is characteristic of carburized gears and confirms the variation in expansion rates given in Table 1. It tends to relieve the tooth load at the tip of the gears and is not detrimental to their operation if not excessive. To compare it with the chart previously given, the expansion rate of this gear tooth is −0.0006.



■ Fig. 2 – Impeller shaft gear tooth expansion during carburizing and hardening

The gear on the right was handled by the same procedure but showed entirely different expansion characteristics. This, we believe, was due to the change in section and the resultant change in cooling rate. A fixture designed to quench the tooth elements of this part at the same rate

as the other should change its expansion rate from —0.0020 to —0.0006. While there is a difference in material, one being AMS 6240 and the other 6294, this is not believed to influence the rate materially. We have other gears of these two materials which do not show similar differences in expansion.

This second gear is at present being finished on a grinder.

Preliminary data collected for the development of shaving cutter forms show a range of rates from —0.0003 to —0.0012 on the teeth of six different gears investigated. The sections of these gears are comparable to that shown at the left on Fig. 2.

▪ Development of Fillet Contours

Another objection to shaving frequently expressed is the probability of highly concentrated stresses due to cutter marks and the line of demarcation between the hobbed and shaved contour in the tooth fillet.

Early in our experience with shaved gears, we were called upon to increase the strength of a 14-tooth pinion driving a 33-tooth gear. Since a higher gear ratio was also desired, we substituted a 13-tooth pinion cut out of the same blank and mating with the same 33-tooth gear. In addition to getting the increased strength and ratio desired, an unanticipated advantage was obtained. The heavy tip contact we had been perpetually fighting on shaved gears virtually disappeared.

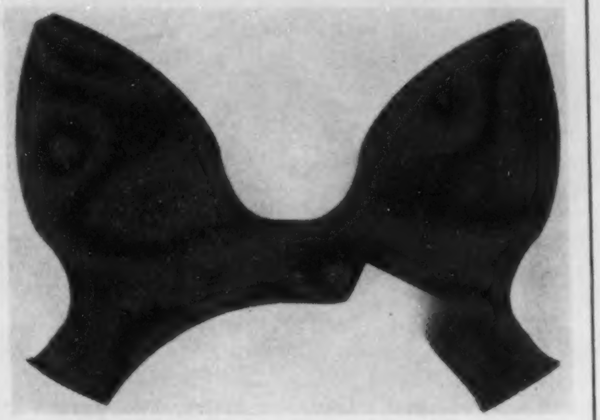
The subsequent Chevrolet transmissions designed to be shaved incorporated dual pressure angles on the gears, a generating pressure angle, and a higher operating pressure angle. These changes permitted the involute form to be generated lower than the mating gear contacted without increasing the depth of the tooth. However, it did decrease the strength of the tooth by narrowing the base. This was compensated for by putting a large radius, tangent to both sides, on the tip of the hob. The result was a slightly increased depth of tooth, but a stronger tooth than one designed by the usual procedure, due to the fact that the radius on the tip of the hob reduced the stress concentration at the base of the tooth.

This procedure was used in laying out hobs to be used prior to shaving of aircraft-engine gears. While it is not a highly stressed gear, the 7-tooth oil pump gear was laid out in this manner and will serve as an illustration of the method. This is a common design of gear and, incidentally, would interchange with a Chevrolet automobile oil pump gear except for the bore and face width. The tooth has been made shallower than standard to permit the use of a larger shaft. The nominal clearance over the tip of the mating gear is only 0.004 in.

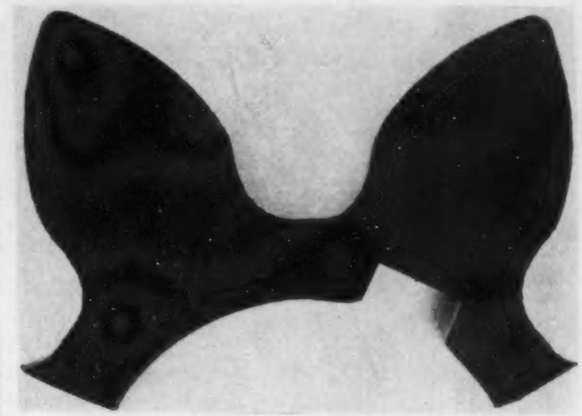
A certain 6-pitch gear of 28-deg pressure angle originally was ground with a rack-shaped grinding wheel. The form of the tooth obtained is shown in Fig. 3. Fig. 4 shows the form of the tooth which would be obtained using a hob of the same form as the grinding wheel. Due to this hob having only 12 teeth, the steps in the tooth fillet are of considerable height. Fig. 5 shows the form of tooth laid out for hobbing prior to shaving to get the maximum strength and smoothest fillet possible. This form is developed by a 20-deg pressure-angle hob having the same normal pitch of 0.4623 as the 28-deg pressure-angle hob. To accomplish this change in pressure angle, the diametral pitch is changed from 6 to 6.3856. The radius formed on the tip of the hob is 0.082 in.

As will be seen in the comparison made in Fig. 6, the resulting tooth form is stronger, as demonstrated by Lewis Y factors of 0.063 and 0.069 for the ground and hobbed forms respectively. The rate at which the tooth form departs from the parabola is in proportion to the stress concentration at the point of tangency. The stress concentration is considerably higher on the form ground with a 28-deg rack-shaped wheel than on that hobbed with the 20-deg special hob.

Six-Pitch Gear of 28-Deg Pressure Angle

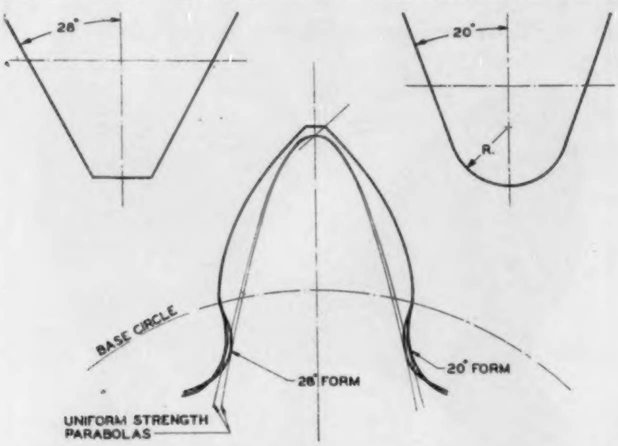


▪ Fig. 3 – Ground with a rack-shaped grinding wheel



▪ Fig. 4 – Form of tooth when using a hob of the same form as the grinding wheel

▪ Fig. 5 – Form of tooth laid out for hobbing prior to shaving to get maximum strength and smoothest fillet possible



■ **Fig. 6 – Comparison of beam strength of teeth generated with 28-deg and special 20-deg racks**

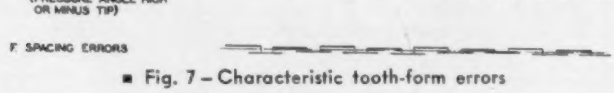
This point of tangency is the only location on the tooth where serious consideration must be given to a smooth contour. Almost without exception, fatigue failures will start very close to the weakest point as indicated by a layout such as this.

The blend of the hobbed and shaved contours is usually smooth enough and far enough away from this point to make special consideration of stress reduction at the blend unnecessary. For the tooth shown in Fig. 6, the blending occurs just below the base circle.

■ Development of Active Tooth Profile

The shaving process by itself does not produce the gear tooth profile as grinding does. It is necessary to control the preliminary hobbed or shaped form to obtain a good shaved gear. In addition to remaining in the fillet and root of the finished gear, this preliminary form also has considerable influence over the active profile which is afterward shaved.

The graphs of Fig. 7 represent the types of error that might occur in the gear before shaving and be only par-



■ **Fig. 7 – Characteristic tooth-form errors**

tially corrected. Success in correcting them in the shaving operation depends on slow cutter feed and rigidity of the gear support. Efforts to eliminate them entirely are best spent on the preliminary operations.

The length of each graph represents one revolution of a 16-tooth gear. It is a combination of the involute profiles and the normal pitch spacing errors which will be found in checking the gear. Upward slopes represent accelera- tion when a standard gear is driven by one having the errors designated. Downward slopes represent deceleration under the same circumstances.

Hob eccentricity (shown in graph *A*), gear eccentricity (shown in graph *C*), and long normal pitch (shown in graph *D*) are the easiest errors to correct in shaving to the point where they are not detrimental to the operation of the gear.

The effects of shaper cutter eccentricity vary according to the portion of the cutter used for making the finish cut on the gear. In extreme cases, it will give a decided spacing error at the point where the cut is completed. Elimination of this error is extremely hard to accomplish.

Long and short normal pitch (shown in graphs *D* and *E*) are due to errors in the original cutters or a change in the cutting angle when resharpening. Short normal pitch is hard to correct by shaving because of the necessity for removing excess material at the root of the gear tooth.

Erratic spacing errors (shown in graph *F*) are due to errors in the cutters and are practically impossible to eliminate completely.

A shaving cutter combines a burnishing and a scraping action. With a low crossed axes angle, the burnishing is predominant. With a high crossed axes angle, the scraping is predominant.

With a pure burnishing action, the amount of material removed at any given point on the profile depends on the number of tooth surfaces in contact at the same time and on the amount of sliding action between the gear and burnisher. At the pitch line, there is pure rolling action so that no material is removed. The sliding action increases with the distance from the pitch line. Burnishing will always leave a hump at the pitch line. This hump can be shifted by changing the burnisher size and altering the pressure angle, but not eliminated.

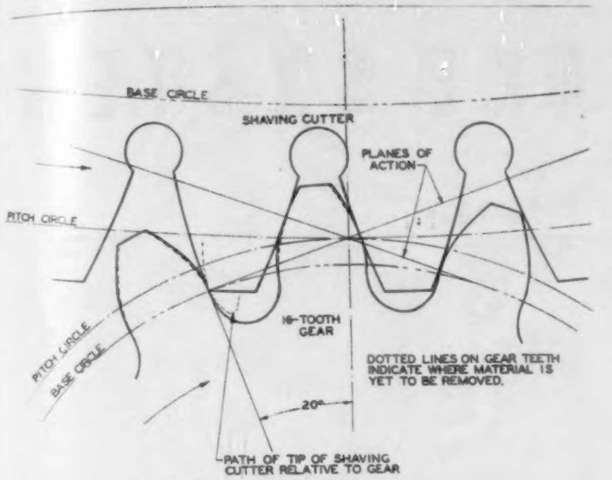
With a rotary-type shaving cutter, an increase in the crossed axes angle increases the curvature of the cutter tooth surface in the plane of action and the unit pressure on the gear tooth. The sliding action lengthwise of the tooth is increased also with both rotary- and rack-type cutters. As this angle increases, so does the cutting action at the pitch line of the gear, provided there are sufficient teeth in contact, until it finally balances the tendency for the burnishing action to leave a hump there.

We then have these variables to influence the cutting action: pressure angle, teeth in contact, and crossed axes angle. The pressure angle varies as the cutter is reground. The others can be constant.

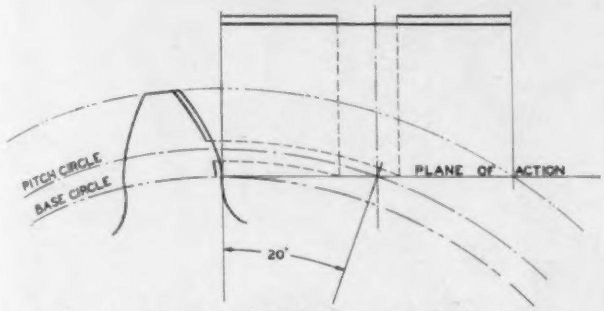
Fig. 8 shows the action of a shaving cutter on a 16-tooth gear in the plane of rotation. The dotted line in the tooth space shows the path of the shaving cutter tip as it comes into contact with the gear. It is this corner which gives the characteristic groove near the base of a burnished or shaved gear. The depth of this groove is reduced by hobbing the involute form deeper than the shaved form, either by the use of lower pressure-angle hobs, as previously mentioned, or protuberance hobs.

Fig. 9 shows the contacts between the teeth of this shaving cutter and gear. These are shown along the plane of action and also along the tooth of the gear. The partial full line near the base circle shows the interval over which there is also contact on the preceding tooth. The partial full line above the pitch circle shows the interval over which there is also contact on the following tooth.

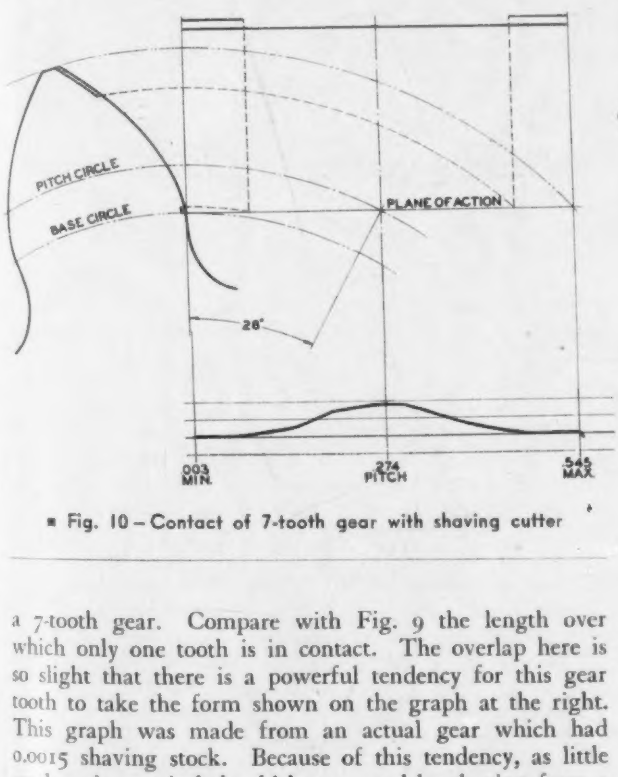
Fig. 10 shows the contacts between a shaving cutter and



■ Fig. 8 – Action of shaving cutter



■ Fig. 9 – Contact of 16-tooth gear with shaving cutter



■ Fig. 10 – Contact of 7-tooth gear with shaving cutter

a 7-tooth gear. Compare with Fig. 9 the length over which only one tooth is in contact. The overlap here is so slight that there is a powerful tendency for this gear tooth to take the form shown on the graph at the right. This graph was made from an actual gear which had 0.0015 shaving stock. Because of this tendency, as little stock as is practical should be removed by shaving from a gear of this type. If the involute form alone is considered, shaving a gear with only slight contact overlap is detrimental; but spacing and normal pitch errors are subject to correction by shaving and can cause serious tip and root interference in operation.

Shaved and Ground Gears Compared

My entire grade school education was under a principal who considered it her duty to encourage right-hand writing. Unfortunately, my natural inclination was to use my left hand wherever there was any option. Recollections of my early school years are full of sore knuckles and overtime sessions to recopy writing with the other hand. However, by the time I reached the fourth grade all methods of persuasion had been given up. In a private session, I was informed that I could write any way I pleased from then on, but the highest mark to be expected for left-hand writing would be 80%.

There seem to be some people like that principal today who consider shaving gears a left-hand method of making them. The process is expected to start off with a 20% handicap. Others consider grinding just as objectionable.

Good finished gears can be made by either method. Both methods have certain inherent advantages that must be made use of, and inherent disadvantages that must be compensated for to obtain the best gears.

Strength and durability are not affected by the method of manufacture, where proper controls are used. Attempts to speed up production of ground gears result in surfaces softened by the heat generated in grinding and the subsequent formation of grinding cracks if carried to an extreme. Luckily, the cracks developed in grinding teeth seldom follow along the base of the tooth. However, we can be quite certain that the heat of grinding has produced surface stresses over that area whenever grinding cracks show up.

Since the quality of ground gears depends so much on the grinder operator, it is possible for such gears to vary from tooth to tooth and gear to gear. At a time when rapid expansion of production is called for and new operators have to be trained, this possible variation puts a heavy responsibility on the gear inspection department. In contrast, the errors due to shaving are most likely to show when the machine is set up on a new gear or cutter. Deterioration of the cutter is very gradual so that set-up and periodic checks are sufficient to control the form of the shaved gear. However, successful shaving does depend on a blank that is more closely controlled than is required for grinding. This tolerance in favor of grinding is not as great as might be at first supposed, because a variation in grinding stock affects the final case depth and the speed at which the grinding operation can be performed without danger of checking.

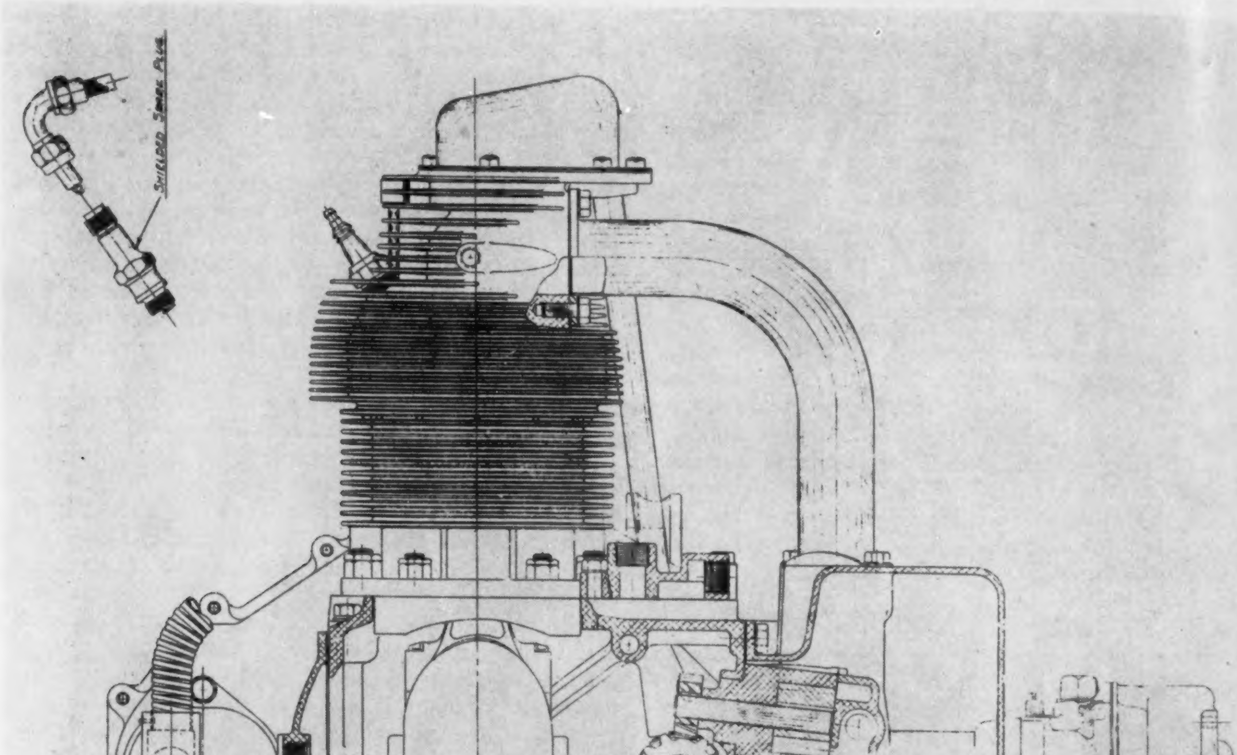
Changes in material such as might be required in the present emergency are easier to incorporate when grinding is used than when shaving is used for the finishing operation. Revision of the shaving cutter form may be necessary where no change of the grinding wheel form is required. However, a change which is radical enough to affect the cutter will also require considerable testing of the product. During this test period, the necessary changes in the shaving cutter can be established.

Gears Suitable for Shaving

Our experience with shaving leads us to select first, for that process, gears which are small in diameter and have

turn to page 344

INTAKE SYSTEMS



■ Fig. 1 – General arrangement of the horizontally opposed aircooled engine

S for AIRCRAFT ENGINES

by CARL T. DOMAN
Aircooled Motors Corp.

IN contemplating the design of a new aircraft engine, the engineer would naturally prefer to have a definite object, as far as specific output, total weight, and production costs, are concerned. However, based on previous experience, the sales department never lets him operate in this manner. An engine is no sooner on paper, than the sales department is toying with the idea of selling the engine for a different power range. For that reason, if the engineer is honest with himself, he usually will admit in the back of his mind that he is providing enough space in the cylinder centers so that when the pressure comes from the sales department for more power, he can get busy with the reamer and enlarge the bore. This means that until the bore is enlarged, the weight is compromised, also the engine is longer than need be, as designed for a given bore and without considering any later enlargement whatsoever.

The position of the sales department is more or less a natural one, inasmuch as the sales volume of an engine in the lower horsepower bracket has not reached a point where an engine can be designed for a specific speed and output. The management is faced with the problem of taking an engine with a given bore and stroke and operating it at different speeds or with different compression ratios, and thus be in a position to offer a range of power which would meet the demands of various airplane manufacturers without necessitating tooling a broad range of various models.

Going back to the original thought, where the engineer must decide on the horsepower, weight, and cost figures, it is safe to assume that he never pictures actually what the given design may be eventually asked to do, when the sales department presents a picture for an engine of different characteristics. This paper, nevertheless, will attempt to point out the engineer's job in arriving at a design from the basic selection of bore, stroke, number of cylinders, valve size, and so on, as related to the induction system itself. This discussion will be confined principally to a series of horizontally opposed aircooled engines having a basic cylinder size of $4\frac{1}{4}$-in. bore and $3\frac{1}{2}$-in. stroke. Fig. 1 shows the general arrangement of the proposed type of engine, having the bore and stroke mentioned. In arriving at the particular dimensions for this engine, the considerations listed below were followed:

a. Width of the engine must be held not to exceed $30\frac{1}{2}$ in.

b. Engine ultimately must deliver 27 to 33.5 hp per cyl at a speed of 3500 to 4000 rpm and 16.2 to 30 hp at speeds of 2000 to 3300 rpm.

c. It must not exceed a weight figure of 245 lb for the

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 9, 1943.]

THE effect of the aircraft-engine intake system on the design of other engine components is here related by Mr. Doman, who applies his findings particularly to a series of $4\frac{1}{4} \times 3\frac{1}{2}$ horizontally opposed aircooled engines.

Valve sizes were varied to obtain the maximum size of inlet valve without appreciable restriction of the flow through the exhaust valve, and at the same time that would provide a strong bridge in the cylinder head between the valve openings.

Care must be taken that restrictions to flow are not created in the intake port. "The intake port," Mr. Doman says, "is probably the most critical item in the whole induction system as far as affecting power output."

In horizontally opposed engines, the carbureter is usually mounted on the oil pan underneath the engine, despite the objections that the carbureter interferes with the streamline of the undercooling and, if a retractable nose wheel is used, the most suitable location brings the nose wheel into the cowl where the carbureter is located.

One solution has been a "runner" type of manifold, which the author applies to 4-, 6-, and 8-cyl horizontally opposed engines. At present, this type does not give as uniform mixture distribution as the system with individual pipes, but Mr. Doman is confident that this disadvantage will be overcome with more intensive research work.

■ ■ ■

THE AUTHOR: CARL T. DOMAN (M '26), vice-president and chief engineer, Aircooled Motors Corp., and for several years secretary-treasurer of the SAE Syracuse Section, took his first job with the Franklin Automobile Co. Many of the aircooled engine patents formerly held by the Franklin Automobile Co. and now owned by the Aircooled Motors Corp., are the inventions of Mr. Doman in collaboration with Edward S. Marks.

■ Fig. 1 – General arrangement of the horizontally opposed aircooled engine

Table 1 – Combinations of Size and Performance That Might Result from a Given Basic Engine Design

Model	Bore	Stroke	Displacement	Rated Hp	Rated Speed	Hp per Cyl	Compression Ratio	Fuel
4AC-176	4	3½	176	65	2300	16.3	6.3	73
4AC-176	4	3½	176	80	2500	20.0	7.0	80
6AC-264	4	3½	264	120	2600	20.0	7.0	80
4AC-199	4¼	3½	199	65	2000	16.3	6.1	73
4AC-199	4¼	3½	199	90	2500	22.5	7.0	80
4ACG-199	4¼	3½	199	108	3500	27.0	7.0	80
6AC-298	4¼	3½	298	130	2550	21.7	7.0	80
6ACG-298	4¼	3½	298	170	3500	28.2	7.0	80
6ACG-298	4¼	3½	298	185	3500	30.8	9.3	100
6ACG-298	4¼	3½	298	200	4000	33.3	9.3	100

Characteristics of Inlet and Exhaust Valves

Model	Bore	Stroke	Displacement per Cyl	Net Area Inlet Valve	Net Area Exhaust Valve	Valve Lift	Displacement per Sq In of Inlet Area	Displacement per Sq In of Exhaust Area	Engine Velocity, 4000 Rpm: Inlet Valve, Fps	Exhaust Valve, Fps	Inlet Manifold Pipe, Fps
4AC-176	4	3½	44	2.126	1.309	0.382	20.69	33.62	230	373	257
4AC-199	4¼	3½	50	2.126	1.309	0.382	23.52	38.19	260	422	290

4-cyl engine and 322 lb for the 6-cyl engine, completely equipped with magnetos, starter, and generator.

d. The selection of the materials used would be based on the minimum cost with maximum durability or life.

The original study of the cylinder was made on a basis of 4-in. diameter and 3½-in. stroke, and the valve sizes varied to obtain the maximum size of inlet valve without decidedly restricting the exhaust valve and at the same time provide a practical and safe bridge in the cylinder head between the two valves. At the same time, the cylinder centers selected were such that the bore could be increased to 4¼ in. when the aforementioned pressure from the sales department would necessitate increased output for a particular application.

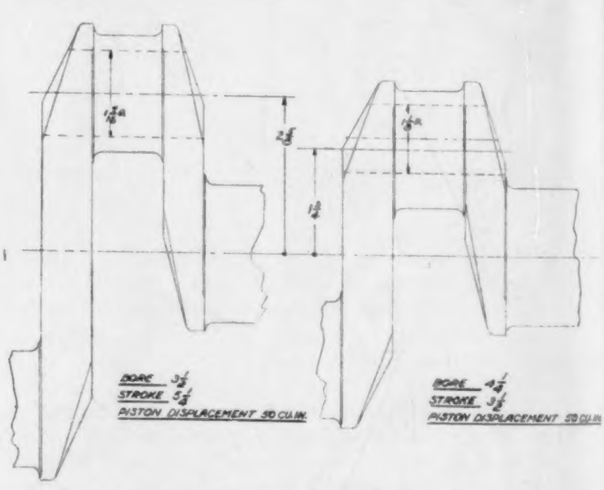
Table 1 shows a summary of the original study of the engine combinations which might result from a given basic design. It is noted that there are two sizes of cylinder bores in both the 4- and 6-cyl models. Also, it is noted that the speed range of these engines is from 2300 to 4000 rpm, with cylinder output of 16.3 hp to 33.3 hp. On the lower part of the table, the characteristics of the inlet and exhaust valves are also given for the two bores with an engine speed of 4000 rpm.

Table 1 also gives the characteristics of the inlet and exhaust valves as related to the size, velocities, and so forth, both for the 4¼-in. bore and the 4-in. bore, for, as mentioned previously, it was constantly kept in mind that the bore would ultimately be increased to 4¼ in.

Of interest, no doubt, is the fact that the inlet valve has 62% greater area than the exhaust valve. Experience has shown that no increase in output has resulted in this cylinder by enlarging the exhaust valve up to ⅛ in. in diameter; in fact, no decrease in output has been found by decreasing the output of the valve below the selected diameter. Of course, at some point the exhaust valve restriction would interfere with the power. Nevertheless, as yet the minimum size permissible has not been determined. On the other hand, an increase in the size of the inlet valve has shown an appreciable increase in power. The inlet valve could not be safely increased in diameter due to the decrease in bridge width between the valves. An increase in the inlet valve diameter resulted in cracking at the bridge. For that reason, the valve size selected represents probably the optimum for the cylinder size selected.

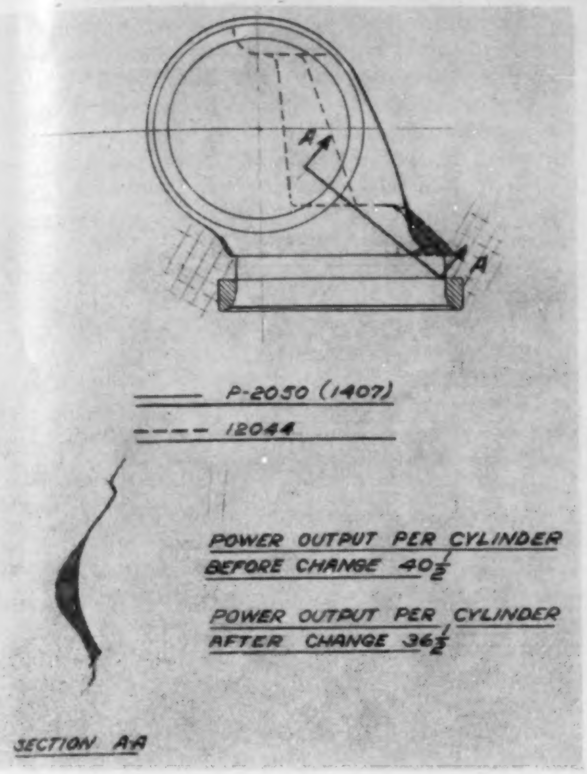
It would be possible, by changing the tooling, to spread the centers of the valves apart on the larger bore cylinder, and thus obtain larger valve diameters. Nevertheless, the design is compromised in order to permit the cylinders to go through the same machining fixtures as used on the 4-in. bore cylinder.

In arriving at the relatively large bore and short stroke, it would appear that the reason for the large bore and short stroke was primarily for keeping the engine width to a minimum. While the narrow width of the engine was important, the primary reason for using the larger bore and short stroke was to provide valves of sufficient size so that the engine would breathe well at the higher speeds. One decided disadvantage of the short-stroke engine is the fact that the crankshaft by necessity is heavier, inasmuch as it isn't possible to use as large lightening holes as in a longer stroke engine. Fig. 2 shows the comparable permissible lightening holes in two cylinders of the same displacement, by using different bore and stroke.



■ Fig. 2 – Comparable permissible lightening holes in two cylinders of the same displacement, by using different bore and stroke – the longer stroke permits the use of larger diameter lightening holes and crankpins

The importance of the intake-port design impresses itself on the engineer only after actual tests have been run with various characteristics. Fig. 3 shows the comparable output with two port designs and the relative difference in designs shaded. Increase in output represents 9.87% at 3200 rpm. While these figures were obtained on a cylinder of 67 cu in., we find they still apply to the cylinder involved in this particular discussion.



■ Fig. 3 – A slight change in the inlet-port design seriously decreases the power output

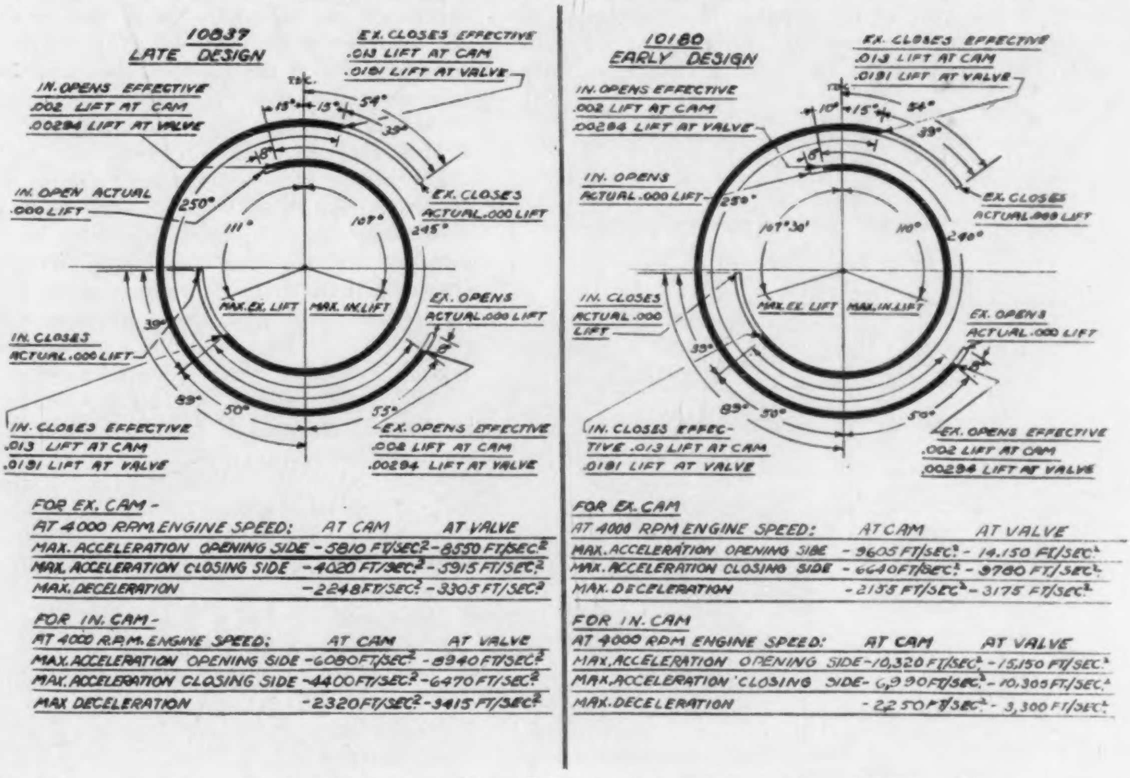
During the development of one of these cylinders, the ports were found to crack where they attached to the combustion-chamber dome. The engineer hurriedly added material to the inside of the port to increase the strength, without checking the flow characteristics of the revised port. It was decided to make several sets of cylinders with the revised port; the change was forgotten in the rush of other work and eventually the engines were tested for power on the dynamometer. The power output was down, and it was only after frantic attempts to find the reason for the low output that the slight increase in metal at the port was found responsible. Flow tests then made showed definitely a restriction had been added. This goes to show that the intake port is probably the most critical item in the whole induction system as far as affecting power output.

In projecting the use of this basic cylinder design over a series of direct-drive engines operating from 2000 rpm to an operating speed of 4000 rpm in the geared type, Fig. 4 shows some very interesting points. Obviously, an engine which would be operated at 2000 rpm would require a different camshaft design than one to operate at 4000 rpm. For that reason, in the first series of engines built, it was decided to use a camshaft which would give satisfactory performance up through 2650 rpm.

An attempt will be made to point out items of design which must be altered to provide best operation in the two speed ranges mentioned. When the basic cylinders were still on the drafting board, and the size of the cylinder had been determined, considerable time was spent in attempting to determine whether to use a valve operating mechanism which was the conventional overhead type, or a mechanism operated through the use of a hydraulic valve lifter. Experience gained in building several thousand air-cooled automobile engines proved it was possible to design an overhead valve mechanism where the change in clearance at the inlet or exhaust valve would not be more than 0.003 hot to cold on either valve.[1] However, the complica-

[1] See SAE Transactions, Vol. 26, pp. 339-345 (also *SAE Journal*, Vol. 28, February, 1931, pp. 177-183): "Development of the Franklin Direct Aircooled Engine," by E. S. Marks and C. T. Doman.

■ Fig. 4 – Camshaft design

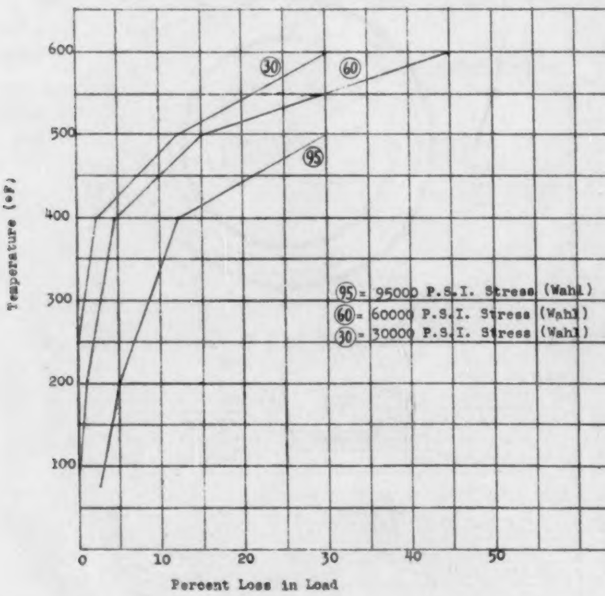


tions of this mechanism and the inability to make an oil-tight mechanism prompted the decision to use the hydraulic valve lifter. Anyone who has used a hydraulic valve lifter for a period of years appreciates the development problems encountered in the valve mechanism in obtaining satisfactory operation from the hydraulic lifter. For that reason, close cooperation must be exercised between the maker of the hydraulic valve lifter, the valve-spring maker, the camshaft designer, the valve maker, the maker of the allied parts, and, of course, the engine designer.

In general, the engine maker obtains usually the best results in the shortest possible time by concentrating, in the hands of the manufacturer of the hydraulic valve lifter, the responsibility for the camshaft design, pushrod design, the rocker arm versus valve stem location, valve spring, the valves, valve guide, and, generally, the design of the valve seats. The reason for concentrating these items in the hands of the hydraulic lifter manufacturer will be discussed briefly as this paper progresses. In addition to the responsibility assumed by the hydraulic lifter manufacturer for parts mentioned above, it also is practical to obtain advice on the design of the valve guide bosses and valve guides, in order to obtain the maximum cooling and also the minimum of valve cocking during the valve operating cycles.

Of major importance in the responsibility assumed by the hydraulic valve lifter manufacturer, is the valve-spring design. The spring must be so designed that it is not overstressed and that it does not have a natural period which will result in valve float and the resulting lifter pump-up. The spring must also use a material which is not seriously affected by heat flowing into the spring from the exhaust port through the spring seat and also must have a range which would permit the engine to be dived 20% to 30% overspeed without lifter pump-up.

The effect of temperatures on the tensile strength of the material is not fully appreciated. For that reason, it behooves the designer of the cylinder head to design the port so that the minimum heat travels onto the valve spring. Reference to Fig. 5 shows a change in valve-spring characteristics with temperature. In other words, the valve spring must be so located and of such a material that it does not "tire" with temperature or life.



■ Fig. 5 – Effect of heat and stress on mean base operating temperatures – Wilcox-Rich curve sheet

At the time the camshaft design is being studied, the decision must be made whether the camshaft will be steel, in which case a hardened cast-iron lifter is usually used, or, if the camshaft is hardened cast iron, a steel lifter is used. Experience learned the hard way has indicated the necessity for following this selection of materials in order to eliminate lifter face failure, or in many cases, failure of the cam nose.

In the usual aircraft design, insufficient thought is given to camshaft stiffness. For that reason, it is well to compute, and if necessary, run tests, to determine the flexing of the camshaft in between the bearings; otherwise false motion will be introduced into the mechanism, and, as a result, the hydraulic lifter mechanism will not function properly.

In general, a larger diameter in the barrel of the camshaft with a large lightening hole gives best results. Of course, again, engine design is a compromise, and maximum camshaft size can only be selected within certain limits.

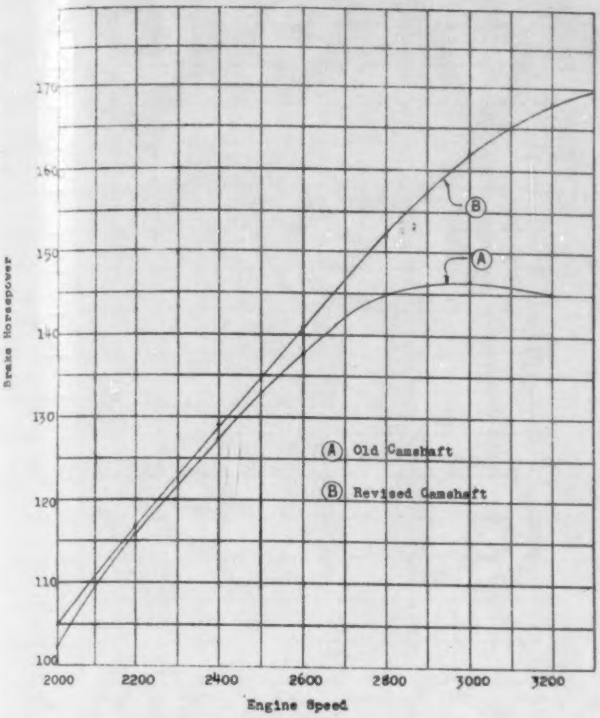
During the development days of the 4-cyl engine, of the 4AC-176 type, it was found that there seemed to be a tendency for the hydraulic valve lifters to pump up at speeds above 2700 rpm. However, at speeds lower than this, no trouble was encountered. Nevertheless, when the 4-cyl engine of the 4¼ x 3½-in type, Model 4AC-199, entered production, it was found that at 2450 rpm, very erratic operation existed in certain engines. Increasing the valve-spring tension did not seem always to eliminate the trouble. However, it was found that if the lubricating oil was extremely hot, the floating condition was at a minimum. For that reason, investigation of the leak-down rate of the hydraulic valve lifters showed that if the rate were kept at 2 to 5 sec, this erratic operation was eliminated, whereas if the leak-down range was between 6 and 14 sec, the lifters would often pump up and the engine would operate in much the same manner as with a "hunting" governor. In other words, the engine speed would oscillate, so to speak, between 2300 and 2500 rpm.

Specifying the lower leak-down rate and accepting the slight increase in valve noise during idling, permitted production to go ahead. Nevertheless, the final solution in eliminating the pump-up at 2450 rpm was the use of a camshaft design as shown on Fig. 4. While this camshaft showed a slight decrease in power at the rated speed (90 hp at 2500 rpm), actually the operation was much more satisfactory. The engine operated much more quietly and continued to operate for a much longer period of time between valve regrindings. This was due to the fact that the valves were not held off their seats by false motion in the valve mechanism.

It was found that with this change in camshaft design and with no other changes, it was possible to operate this same engine up to slightly beyond 4000 rpm without any apparent erratic operation in the valve mechanism.

Fig. 6 shows an unexpected result in that in the 6-cyl engine of the same bore and stroke, the power output is generally higher with the revised camshaft, possibly due to the more definite operation of the valves. However, this does not seem the logical explanation, in that the same camshaft design in the 4-cyl engine showed a slight decrease in power.

Inasmuch as the aircraft engines in the lower horse-



▪ Fig. 6 – Power output of a 6-cyl engine with the old and the revised camshafts

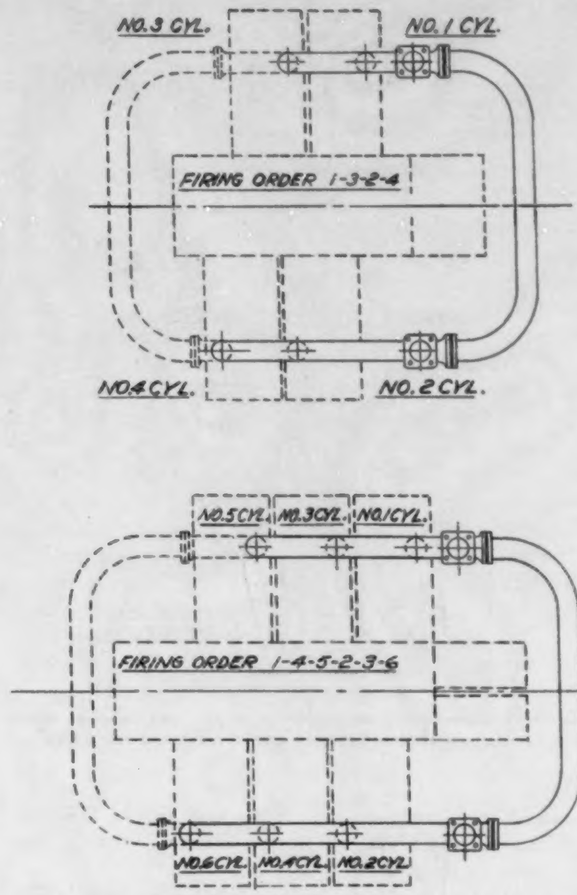
power bracket are all of the horizontally opposed type, it is usual practice, for simplicity and for cost purposes, to mount the carbureter underneath the engine, as shown in Fig. 1. The reason for this more or less universal location of the carbureter is evident. Nevertheless, it might be well to repeat the reasons for this location. They are as follows:

1. Eliminates the necessity for a fuel pump with possibility of failure.

2. The carbureter riser is usually cast in the oil pan and receives heat from the oil for better vaporization when cold and also acts as an oil cooler.

3. This location permits the use of a long, individual pipe from the distributing zones, which gives an inherent supercharging effect.

There are, of course, several objections to the use of the carbureter on the oil pan, principally due to the fact that the carbureter interferes with the streamline under-cowling,



▪ Fig. 8 – Diagrammatic arrangement of the runner type of manifold for a 4-cyl engine (above) and a 6-cyl engine (below).

inasmuch as the carbureter, or airscoop, hangs out of the bottom of the cowl. Also, if a retractable nose wheel is used, the designer usually wants to pull the wheel up into the cowl at the exact usual location for the carbureter, or carbureters.

To eliminate these two objections outlined, a so-called "runner" type of manifold has been developed, which has been found to be very satisfactory up to 3500 rpm. Fig. 8 shows diagrammatically this type of manifold as applied to a 4-cyl and a 6-cyl engine, while Fig. 7 shows this type of "runner" manifold on a 6-cyl engine. For reference purposes, Fig. 9 shows diagrammatically conventional manifolds used on 4- and 6-cyl opposed engines. These are so-called "pan" type manifolds.

It is noted that with this general manifold arrangement on both the 4- and 6-cyl engines, two carbureters are used. The reason for using dual carbureters on the 6-cyl engine, at least,

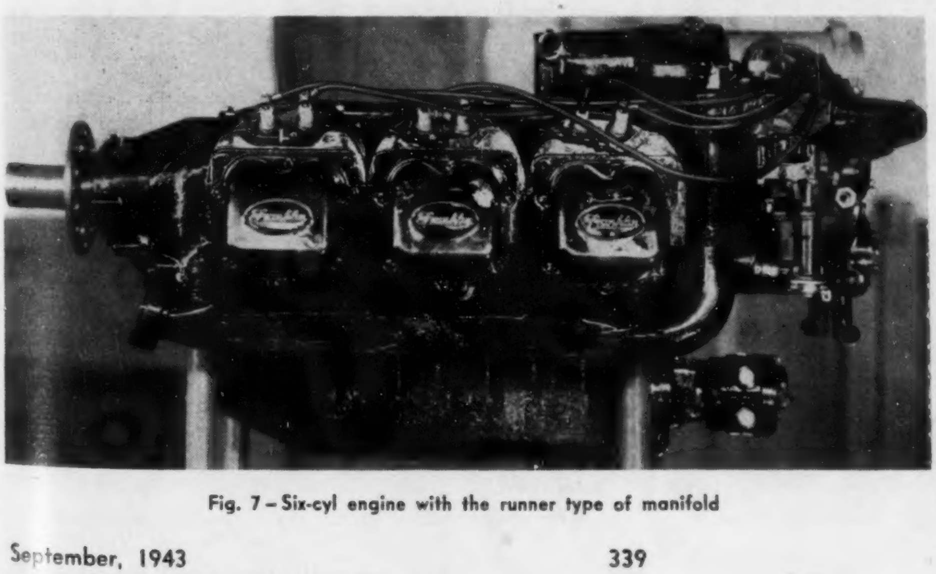
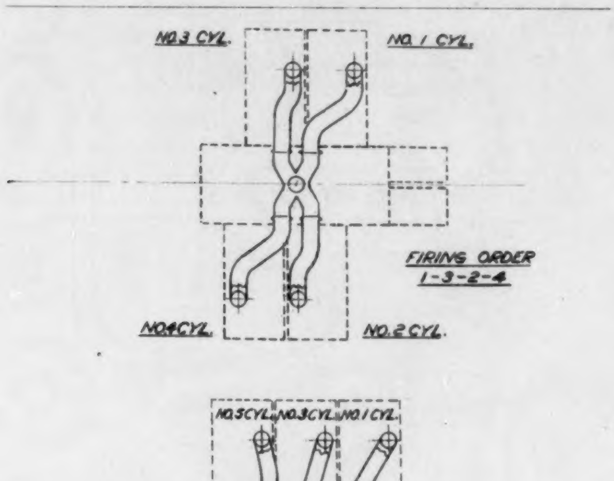
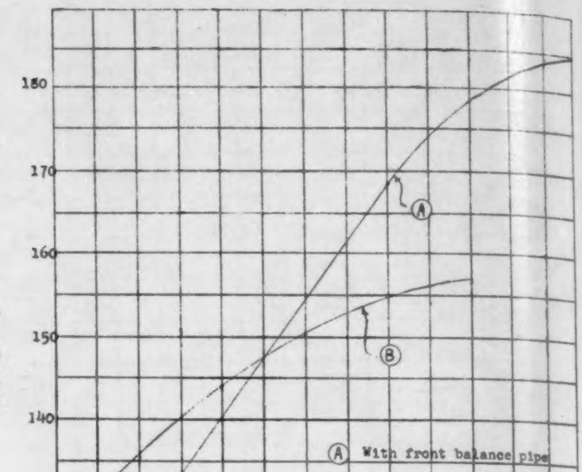


Fig. 7 – Six-cyl engine with the runner type of manifold



■ Fig. 9 – Diagrammatic arrangement of the pan type of manifold for a 4-cyl engine (above) and a 6-cyl engine (below).



■ Fig. 10 – Power output of a 6-cyl engine with and without front balance pipes

is covered in a previous paper.[2] In that article, however, there was no discussion of the possibility of using dual carbureters on the 4-cyl engine.

In developing this type of manifold, poor peak power resulted; it was found that by the use of a front balance pipe, the shape of the power curve was entirely altered. Fig. 10 shows the power output of the 6-cyl engine with and without the front balance pipe. It is noted that up to 2700 rpm, the power output is decidedly better without the front balance pipe. However, for speeds above this, the power is increased from the use of the pipe, for example, by 21 hp at 3200 rpm.

In actual flight conditions in a relatively fast plane, with a constant-pitch wooden propeller, it has been found that without the front balance pipe, the climb is much better, whereas at full-out level flight, speed is sacrificed. This is explained, of course, by the fact that if the engine is turning at 2200 rpm during a climb, it may level off at 2800 rpm under full-throttle level-flight conditions. At 2200 rpm, it would have available for take-off with a balance pipe 112 hp and without the balance pipe, 126 hp.

One other advantage, not as yet mentioned, with the runner type of manifold with the carbureters located alongside the rear cylinders, is the possibility of mounting the engine in the wing, as has been proposed by many who perhaps are better acquainted with the drafting room side of an airplane than they are with the actual flight characteristics of the airplane.

No particular mention has been made of the use of dual carbureters. However, it has been found that on the 6-cyl engines, brieflly described in this discussion, the power output with a single carbureter for speeds above 3000 rpm has never approached that obtained with dual carbureters. It is appreciated that dual carbureters do incorporate certain complications of hook-up. Furthermore, if the gas supply to the carbureter becomes clogged, the engine suddenly stops firing, inasmuch as air is drawn through that carbureter and the mixture leaned up to a point where the engine will not operate.

It is hoped that the carbureter manufacturers will see their way clear to develop a dual carbureter having one float, or of a design following the same general features of the popular dual automotive carbureter. It need not matter whether it is up-draft or down-draft as the engine manufacturer could readily alter the design to suit the particular application.

The work incorporating dual carbureters in the 4-cyl engine has been quite limited. However, reference to Fig. 11 shows a decided gain in power at 2400 to 3400 with the dual carbureters located on the oil pan. However, the increase from dual carbureters when using the runner-type manifolds (see Fig. 12), is not as great. It is to be noted that less power is obtained up to 2730 rpm with dual carbureters than with single.

Mention has been made previously of the supercharger effect from the use of the long intake pipes. It has been found that the compression pressures have actually been boosted 30 psi by the use of these long individual pipes from the riser. Table 2 shows the compression pressures at 2750 rpm with and without the intake pipes connected, as measured on a 6-cyl engine.

It is felt there is a great deal of work to be done on the selection of pipe sizes and lengths. Nevertheless, usually an engineer is limited by the space available underneath the engine. Experience has not shown that the pipe is not tuned for any particular speed, as the increase in output from the long pipes has been found to be quite uniform over a broad range of speed; that is, from 2200 rpm to 3200 rpm.

The intake system having the individual pipes from the

[2] See SAE Transactions, Vol. 50, May, 1942, pp. 188-195: "Light Plane Engines and Their Fuel Problems," by Carl T. Doman.

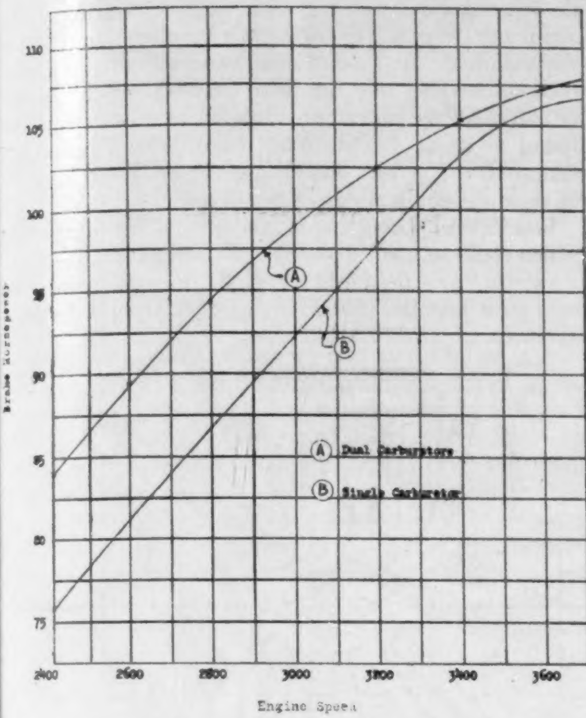


Fig. 11 – Performance of a 4-cyl engine with single and dual carbureters located in the oil sump

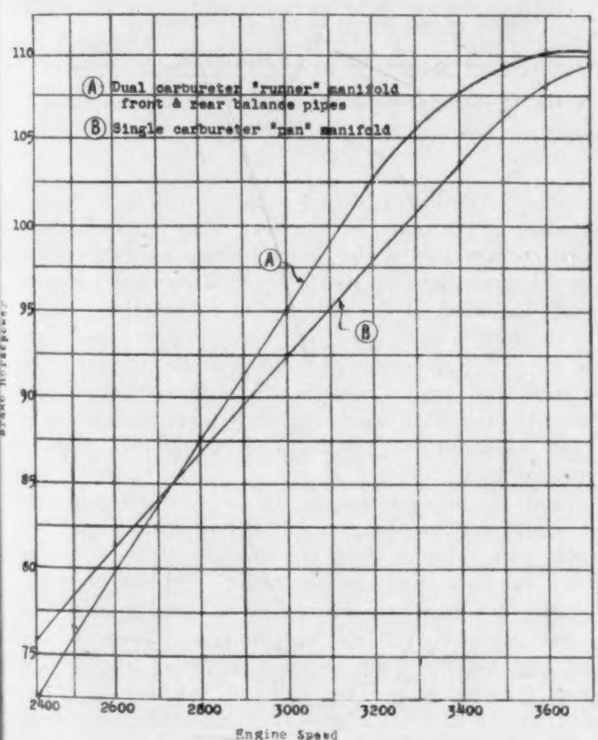


Fig. 12 – Increase in output obtained from the use of dual carbureters, with the runner type of manifold, on a 4-cyl engine

riser to each of the cylinders also gives fairly good mixture distribution, both on the 4-cyl as well as the 6-cyl engine. Usually, a maximum spread of 2 ratios is found on either the 4- or 6-cyl engines.

Thus far, the runner type of manifold, as shown in Fig.

Table 2 – Effect of Long Induction Pipes on Compression Pressure

Rpm	Compression Pressure, psi, Motoring Cylinders 1	2	3	4	5	6	
2750	175	204	182	198	212	194	With intake pipes
	157	160	163	157	162	148	Without intake pipes
	180	178	180	172	188	146	With intake pipes and no carbureter

Tests taken on Model 6AC-264 engine – 6 cyl opposed, 4-in. bore, 3½-in. stroke

7, has not given as uniform mixture distribution as the system with individual pipes to each port. This, no doubt, is explained by the fact that the runner type of manifold has not had as intensive work conducted on it as the very popular type with the individual pipes to each cylinder. There are indications that in extremely cold weather operation, without manifold heat, the operation becomes somewhat unstable, whereas with the system having the individual pipes, the performance is very constant, regardless of temperatures. It is quite remarkable, nevertheless, that either system will operate as well as it does without anv heat whatsoever at the zone. It has been found that with the runner type of manifold, the application of even a small amount of heat by circulating the hot oil from the lubricating system around the zone, will tend to make the system much more stable.

In selecting the spark-plug location, of course the engineer would like to locate the plugs for maximum combustion efficiency. However, in many cases he is forced to locate the plugs at a position on the cylinder which is readily accessible, at a sacrifice in efficiency. For easy access, the plugs located on the top of the cylinder are most desirable. However, from an efficiency standpoint, the plugs diagonally opposite give the best results. Fig. 13 shows the power output with various plug locations in a given cylinder. It is noted that with the plugs located on the top of the cylinder, a decrease of 4.2% results, as compared with the plugs located diagonally opposite.

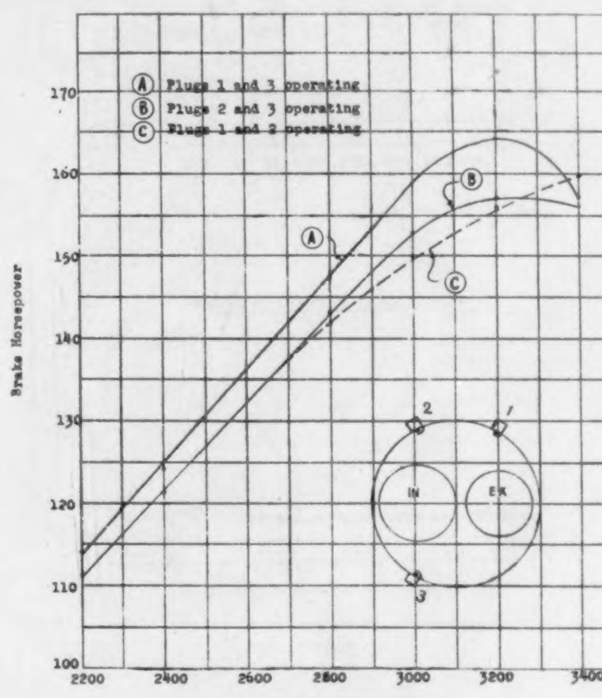
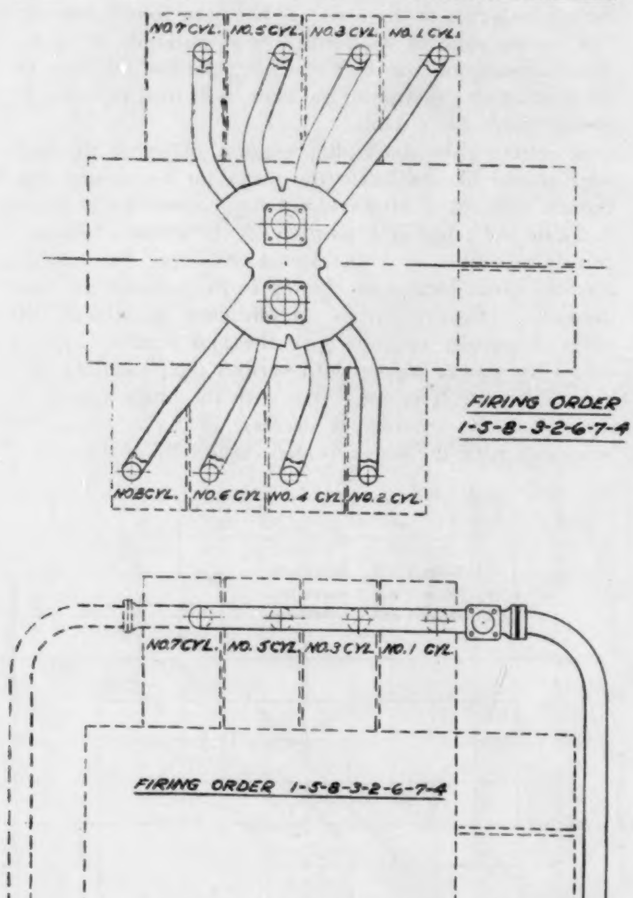


Fig. 13 – Power output with various spark-plug locations in a given cylinder on a 6-cyl engine

In selecting the firing order for a 4-cyl opposed engine, it is customary to follow the arrangement shown in Fig. 8 and in the 6-cyl engine, it is usual practice to follow the arrangement also shown in Fig. 8. It is noted that in the 4-cyl engine, on a given bank of cylinders, two cylinders follow consecutively, whereas on the 6-cyl opposed engine, the firing alternates across from one bank to the other. From a power standpoint, with individual pipes from the distributing zone, the firing order apparently does not affect the output whatsoever.

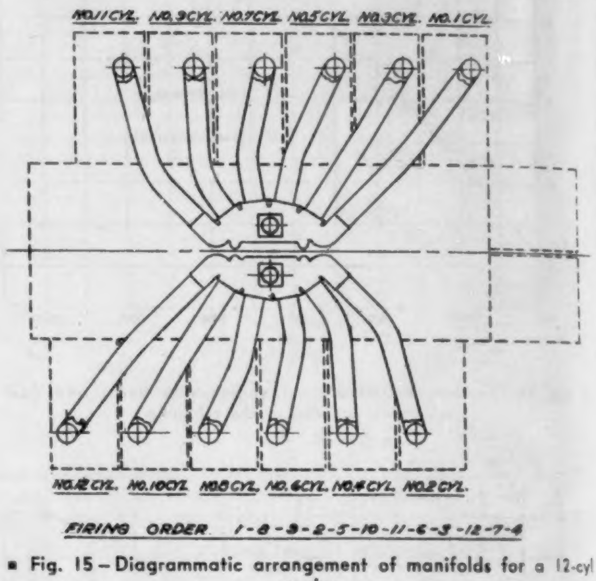
The problem of selecting the firing order for an 8-cyl opposed engine so that it will have 90 deg between explosions is much more difficult than with the 4- or 6-cyl opposed engines. Successful 8-cyl engines were built with two cylinders firing simultaneously. However, these engines were inclined to be rough, inasmuch as the intervals between explosions were 180 deg. A successful firing order for an 8-cyl opposed engine is 1 – 5 – 8 – 3 – 2 – 6 – 7 – 4. (See Fig. 14.)



▪ Fig. 14 – Diagrammatic arrangement of manifolds for an 8-cyl engine

The intake manifold required with this firing order is one which on paper seems to be quite complex, but actually on test engines it has proved to be very satisfactory. We again refer you to Fig. 14 for the general arrangement of the manifold. It is noted that two carbureters are used, both discharging into the same chamber. With this type of manifold, the maximum variation in mixture ratio was found to be 2. On the other hand, with a runner-type manifold, the power output was 20 hp lower; also, the mixture distribution was rather poor.

In a 12-cyl opposed engine, there are many firing orders which could be used. However, the firing order 1 – 8 – 9 – 2 – 5 – 10 – 11 – 6 – 3 – 12 – 7 – 4 is the one which has been most popular. (See Fig. 15.) This firing order neces-



▪ Fig. 15 – Diagrammatic arrangement of manifolds for a 12-cyl engine

sitates a crankshaft design which gives high crankcase loading at the center of the case. This is due to the fact that concentrated at the center bearing are two crankpins in the same plane on which are mounted four connecting rods, inasmuch as the rods are, of course, side by side or of the forked type. To offset this condition, it is necessary to use very large counterweights at the center of the shaft, and in some cases it is impossible within the space limitations to provide sufficient counterweights to eliminate possible overloading of the crankcase metal itself from centrifugal loads.

With the firing order for the 12-cyl engine mentioned, it was found that the principle of two distinct 6-cyl manifolds, each cylinder being fed through an individual pipe from the zone, was very successful. The use of two carbureters for each 6-cyl manifold was found to increase the power approximately 4% at 3200 rpm. However, it was felt that for the slight increase in power, the complications involved would not make it worthwhile. Various attempts to match the power output with the manifold with individual pipes by the use of runner manifolds or carbureters located alongside the engine, have not been at all successful.

An alternate firing order of 1 – 10 – 5 – 2 – 9 – 6 – 4 – 11 8 – 3 – 12 – 7 has been used. With this firing order, the resulting crankshaft is such that the center crank throws are 60 deg apart, resulting in the load of the center of the case being reduced approximately 40% over the conventional 12-cyl crankshaft. On the other hand, the intake

manifolding becomes more of a problem in that on each bank of cylinders are two cylinders which would fire consecutively, as in the right bank 6 and 4 fire consecutively, and then on the left bank 7 and 1 fire. However, it would appear that by balancing them across the two manifolds, satisfactory performance would result.

It is of interest to note that in an experimental group of engines, all utilizing the same cylinder, the power output per cylinder in the 6-, 8-, and 12-cyl models is approximately the same at the rated speed of 3200 rpm, indicating a uniformity in the induction system, even though the firing intervals between successive cylinders might not be as desired.

When this same group of experimental engines is supercharged, it was found that the power output per cylinder for a given manifold pressure is approximately the same for the 6-, 8-, and 12-cyl engines. In the case of the 8- and 12-cyl engines, the manifolding which gave best results from the power, as well as the mixture distribution standpoint, was that incorporating individual pipes from the distributing zone. In the case of the 6-cyl engine, however, the runner type of manifolds with a balance pipe connecting them at the front on the entrance to the supercharger at each end, peculiarly, gave better results than manifolding incorporating individual pipes from the distributing zone.

This paper on horizontally opposed engines having 4-, 6-, 8-, and 12-cyl, has discussed only intake manifolds and allied parts. No mention has been made of the 2-cyl opposed engine, which has in the past been a popular light plane engine, both in the United States and in Europe. Investigation of this type of engine has shown that the same general principles which have been held for engines with a large number of cylinders, also holds, provided the crankshaft arrangement is such that there is 360 deg between explosions. If more than one carbureter is used on the 2-cyl engine, it has been noticed that, in many cases, blow-back through the carbureter is excessive, with the result that the fire problem is one which might cause great concern.

As far as general engine smoothness is concerned, the 2-cyl engine operating at 3500 to 4000 rpm would be perfectly satisfactory for a light airplane, provided the engine is properly installed on the airplane mount with properly designed rubber bushings.

In conclusion, the author wants to point out that there has been no mention made in this paper of the theoretical approach in the design of the various manifolds. Theoretical approach must always accompany such an investigation. Nevertheless, in a paper of this length, it is not felt that space would permit including the theoretical side of the picture.

Chemical Problems of Engine Lubrication: the Problem of Lubricating Oil Stability

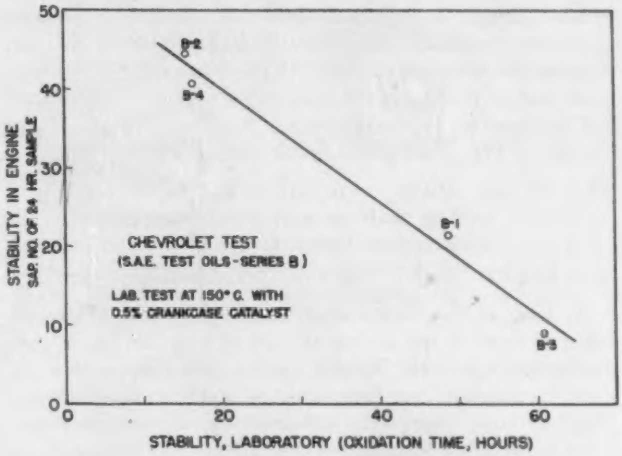
continued from page 316

consumed during the oxidation. In the top curve are given data on the B-1 to B-4 oils of the SAE testing program on the Chevrolet. These oils had been examined by a number of other laboratory tests, chiefly those involving the use of solid-metal catalysts, and almost invariably the oil B-1 had been out of line with engine results. Use of crankcase catalyst brings it into excellent agreement with the engine. In the lower curve are included the data for a number of aviation oils representing both doped and undoped types. Again, good correlation with the engine is obtained. (In both these sets of experiments, the crankcase catalyst used was coarser, and hence less active, than that of Table 1.) Tests on these oils made in the presence of metallic copper show the erroneous rating which may be obtained when sole reliance is placed on experiments using a purely arbitrary type and amount of catalyst. Since crankcase catalyst is variable in composition, it may be preferable to make a synthetic crankcase catalyst using pure compounds.

While the indications described in this report should not be extended beyond the facts which they represent, it is hoped that a new viewpoint has been described which will be of assistance in understanding the complex problems of engine lubrication.

Acknowledgment

The authors acknowledge with gratitude the assistance given by F. J. Watson for making the separations of crankcase catalyst; W. M. Malott and W. Gasser of the Shell Development Motor Laboratories for the engine data; and L. M. Smithbauer for assistance in carrying out the oxidation tests.



■ Fig. 6 – The stability of oils in laboratory and engine tests

The Possibilities of Shaved Gears for Aircraft Engines

continued from page 333

relatively heavy web sections centered under the tooth. The largest gears we are now shaving or are planning to shave in the near future are the 26-tooth gears shown in Fig. 2. Those in the process of development have 14 and 16 teeth. The larger gears with light sections that approach the designer's ideal of a halo mounted on a spider web are being left for future consideration. However, we are shaving some large involute splined couplings.

Small stub-tooth gears having less than 1.3 teeth in contact are not suitable for shaving, if maintaining a true involute form is important. To permit shaving of this type of tooth, some manufacturers have left excess stock on the top of the teeth during the shaving operation and removed it afterward.

It is frequently necessary to change the operations performed after hardening considerably when shaving is introduced. The bearings must then be ground in relation to the gear teeth instead of grinding the teeth in relation to the bearings.

Shaving is a relatively high-production operation which requires controls most readily applied where there is a steady flow of material. If the shaving machine set-up must be frequently changed for different jobs, consistent results cannot be expected. Uniform conditions of heat-treatment necessary for close expansion control are also maintained with greater ease if the equipment is in constant use at the desired temperatures.

Comparative production rates for finishing the gears shown in Fig. 2 are given below in pieces per hour:

Shaved Gear	Shave	20 on Rotary Shaver
	Lap teeth	30 on Internal Lapper
	Recenter	10 on Grinder
Ground Gear	Grind	2.8 on Gear Grinder

As long as the requirement for these gears is below 2.8 per hr, there is no saving in equipment, but as it gets above that figure, the shaving operation begins to show up more favorably. Since the shaved gear operations are being done without using the full capacity of the machines, there is probably considerable improvement yet possible by revision of the tooling. The following production figures for the automotive transmission gears shown in Fig. 1 give some idea of what can be obtained from shaving:

Clutch Gear	63 on Rack Shaver
First and Reverse Gear	67 on Rotary Shaver
Second-Speed Gear	60 on Rack Shaver
Countergear – Front	70 on Rack Shaver
Center	48 on Rotary Shaver
Rear	70 on Rack Shaver
Idler Gear – both ends	44 on Rotary Shaver

Our selection of gears to be shaved was made on the basis of quantity requirements and rigid sections. There are 10 oil pump gears used to each engine which vary only in length and bore size. These variations can be taken care of by arbor changes and do not require any changes in machine set-up. The other gears being considered can be produced in some quantity; three similar gears in each case can be made with the same hobs and shaving cutters.

■ Conclusion

Most of the data given here are based on experience with automobile transmission gears, but we have no reason to believe they will not apply equally well to aircraft-engine gears. Our test experience on the shaved impeller shaft gear shown in Fig. 2 (left-hand) has not extended far enough to predicate the life of the gears. One gear has completed three model tests and part of a fourth. Another has completed a model test and an additional 100-hr durability test at take-off power and speed. Several others have completed single model tests. None of these gears have shown any indication of failure.

Hobs only have been considered for preliminary operations where grinding is being replaced. Shaving can be used after shaping operations as well as after hobbing operations, but the fillet form control is not as easy with a shaper cutter as with a hob.

In the discussion of cutter action, the rotary type of shaver has been considered almost exclusively because the first cost of these cutters is much lower for each application than the rack type. The main advantage of the rack type of cutter is in the production of helical gears, which are not common in aircraft engines.

This discussion is made possible only through the generous cooperation given by many members of the organization of which the author is a member. Particular acknowledgment is made of the assistance given by John Kent, chief draftsman of the Chevrolet Aviation-Engine Plant, in making the illustrations used.

Fuel Requirements for Tanks

MANY tanks are powered with radial motors of the airplane type. These tanks are heavily armored, and to protect personnel, motors, and other equipment in these tanks, little ventilation could be provided. These tanks must carry various types of equipment, armament, and munitions; therefore, limited space precludes the normal spacing between gasoline, supply tanks, and motors. The extreme heat generated in these tanks caused vapor locks when certain types of fuel were used. Correcting for vapor locks caused starting difficulties; therefore the 80 octane, all-purpose gasoline is really the best compromise that could be made on a fuel that would perform satisfactorily in all ground equipment. The Ordnance Department is making experiments, and hopes to write a new specification for an 80 octane, all-purpose gasoline that will, in principle, combine the A and B grades into one grade. The purpose of these experiments is to attempt to adopt specifications for an 80 octane gasoline that will eliminate starting difficulties, and at the same time prevent vapor lock. One of the difficulties encountered is that the supply of this type of gasoline may be considerably reduced by the adoption of such specification. Here is an opportunity for engineers of the oil industry to lend a helping hand, and I am glad to learn that a number of them are now giving this problem their undivided attention.

Excerpts from the paper: "Army Requirements of Fuels and Lubricants," by Brig.-Gen. Walter B. Pyron, chairman, War Department Committee on Liquid Fuels and Lubricants, presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.

LUBRICANTS for Ordnance Combat and Motor-Transport Vehicles

by MAJOR R. E. JEFFREY, Jr.

Ordnance Department

THE general subject of standardization and specifications is not a particularly glamorous one; however, it is not my intent to discuss this subject only, but also some of the development activities that have led to present standard Army lubricants. Judging from the number of questions that have been asked lately regarding these items and activities, it may be well to take this opportunity to trace the course of events leading to the development and standardization of the present types and grades of lubricants.

Standardization of lubricants for the Army has had a very important bearing on military operations, as well as a definite influence on commercial activities during the emergency. Specifications, of course, are the means by which the Army describes the lubricants it requires, and are the means by which procurement of these materials is accomplished. Development of lubricants for military service must finally result in specifications for them, or in revisions of the appropriate ones, in order to be effective.

A word on specification types used by the Army may be of interest before going further. Specifications for lubricants used by the Army are generally in the form of U. S. Army specifications of the 2- series. Thus, engine oil is covered by U. S. Army Specification 2-104. The first major revision would be numbered 2-104A. If a minor revision was next made, the specification would be numbered 2-104A, Amendment No. 1. The second major revision would be 2-104B, and so on. In many cases, when a new automotive lubricant has been developed, and frequent changes are anticipated, it may be covered initially by an Ordnance Department tentative specification of the AXS

[This paper was presented at the SAE Diesel-Engine and Fuels & Lubricants Meeting, Cleveland, Ohio, June 3, 1943.]

★ ★ ★

WHEN the Ordnance lubrication standardization program was started about two and one-half years ago, there were no standard lubricants for military vehicles; consequently, a wide variety of types and grades found their way into Army vehicles, complicating the supply problem and sometimes even resulting in the application of the wrong lubricant.

There were seven types and 22 grades of lubricants for automotive equipment, not including those for special purposes. The first step was to describe, wherever possible, each type and grade of lubricant in terms of some Federal specification already in existence. The problem then resolved itself into:

1. Establishing and maintaining an efficient system of lubrication instructions for issuance to troops.

2. Reducing the number of types and grades to the minimum consistent with satisfactory performance.

3. Developing these types and grades to fit most satisfactorily the military applications involved.

4. Developing satisfactory U. S. Army specifications to cover adequately the necessary materials.

Major Jeffrey discusses the latter three considerations.

In connection with the work on engine oil and gear lubricants, it was necessary to consider low-temperature applications, particularly pumpability, to facilitate cold-weather starting and warm-up.

Greases and engine preservative oils were also studied, resulting in improvements in both. The developments in greases were made particularly to give better cold-weather operation. One of the main problems in respect to preservative oils was that of reducing the number of critical materials. The new specification covers a preservative oil that gives adequate lubrication without being wasteful of critical materials.

THE AUTHOR: MAJOR R. E. JEFFREY, JR. (SM '42), having been an Ordnance reserve officer since graduation from R.O.T.C. at Stanford, was called to service relatively early – in the fall of 1940. He was assigned to the Office of the Chief of Ordnance in connection with the starting of the fuels and lubricants program, involving the standardization and development of these materials, as well as the establishment of a uniform system of lubrication instructions. Major Jeffrey is now chief of the Fuels and Lubricants Section, Service Branch, Technical Division, Office of Chief of Ordnance. He graduated from Stanford University in 1934 with the degree of bachelor of arts in engineering, and in 1937 he received the degree of mechanical engineer. At that time, he became a research engineer on the staff of the Motor Laboratory at the Martinez Refinery of the Shell Oil Co., Inc., Calif.

series. When it is considered that the lubricant requires no further immediate changes, the AXS specification will be superseded by a U. S. Army specification. At the beginning of the current lubricant activity in the Army, however, the best available specification of any Federal agency was used to expedite standardization of lubricants, as we shall see.

Two and one-half years ago, there were no standard lubricants for military vehicles. Consequently, a wide variety of types and grades of these materials found their way into tanks, scout cars, trucks, tractors, jeeps, and other items of Army equipment. At that time, the most serious result of this situation was frequent misapplication of type and/or grade of lubricant from the large variety available. It was also apparent that the supply problem would become serious as the Army expanded and as Army units entered more extensively into field maneuvers, and ultimately into combat. Obviously, the fewer the items to be supplied, the more efficient would be the supply system in providing a sufficient quantity of the proper materials for the fields of operations.

The Ordnance Department recognized the problem at that time, and initiated steps to solve it. In order to standardize the engine oils then in use, it was decided to make use of the Navy specifications for engine oils, using the 1000 and 3000 series as described in the Navy Bureau of Ships Pamphlet No. 431. The heavy-duty oils required for certain diesel-powered tractors were to be those lubricants which had satisfactorily passed the widely recognized Caterpillar series of engine tests and the similarly well-known General Motors 500-hr Engine Test. Straight-mineral gear oils, required by some manufacturers of military axles, would also conform to the Navy specifications of the 1000 series.

The so-called "truck-duty" hypoid gear lubricants, required in truck hypoid axles, were to be obtained under the proposed Federal Specification VV-L-761. Suitable grease types were selected from the rather broad specifications which were then a part of the Treasury Department Procurement Schedule, General Supplies, Class 14. This first step resulted in describing, wherever possible, each type and grade of lubricant needed, in terms of some Federal agency specification already in existence. The types and grades of lubricants then in usage were as follows:

Type	Grades
Engine oil	SAE 10, 20, 30, 40, 50, 60, 70
Diesel-engine oil	SAE 10, 20, 30
Gear lubricants (straight mineral)	SAE 80, 90, 140, 250
Gear lubricants (truck-duty hypoid)	SAE 80, 90, 140
Grease, chassis	NLGI 0, 1, 2
Grease, water pump	NLGI 4
Grease, wheel bearing	NLGI 3

Thus, we see that there were seven types and 22 grades of lubricants for automotive equipment alone, not including those for special purposes.

After the first step, the problem was resolved into four parts:

1. Establishing and maintaining an efficient system of lubrication instructions for issue to troops.
2. Reducing the number of types and grades of lubricants to a minimum consistent with satisfactory performance.
3. Developing these types and grades to fit most satisfactorily the military applications involved.
4. Developing satisfactory U. S. Army specifications to cover adequately the necessary materials.

The first part has been described elsewhere. It is my intention, however, to go into the other three parts of the problem during the course of this paper. Parts 2, 3, and 4 will be covered in separate considerations of each type of lubricant.

Engine Oil

In reducing the number of engine oils, it was first decided to eliminate, as far as military operations were concerned, the intermediate grades, SAE 20, 40, and 70. After further consideration of the engine oil viscosity grade required for aircooled engines used in tanks, it became apparent that the SAE 50 grade could be used in place of both SAE 50 and 60. We then had three standard grades of engine oil, SAE 10, 30, and 50.

As indicated, there was no Governmental specification for diesel-engine oils, and therefore, the first U. S. Army specification written for lubricants in the standardization program was No. 2-104, for this type of oil. This specification was carefully considered by a group representing the petroleum and automotive industries, called together by the Ordnance Department, which met in Detroit on Sept. 3, 1941. Excellent agreement was reached on the specification, and it was published under date of this meeting. The new specification included provision for qualification of these oils prior to procurement, using the same Caterpillar and General Motors engine test procedures noted above. The specification provided for SAE 10 and 30 grades only.

As the military operation of motor vehicles increased both in extent and severity, it developed that the alloy bearings in the engines of many of these vehicles were subject to corrosion with the uninhibited straight-mineral oils then in use. This paralleled commercial experience under similar conditions, and the problem therefore was not difficult of solution. Corrosion inhibitors were known that could be added to the oil. Such inhibitors were already present in the diesel-engine oils being used by the Army, as well as additives intended to impart detergency properties to the oil in order to keep the engine clean. Such detergency properties are also desirable in gasoline engines in heavy-duty service to minimize ring sticking and lacquer formation on piston skirts. It was apparent that if the diesel-engine oil, U. S. Army Specification 2-104, was made the standard engine oil for both gasoline and diesel engines, not only could corrosion of alloy bearings be controlled, but maintenance of the engines could be markedly improved through keeping them clean.

The importance of this, of course, cannot be overestimated. Accordingly, it was decided to standardize the heavy-duty engine oils for all engines of Ordnance vehicles. U. S. Army Specification 2-104 was modified to provide for the SAE 50 grade as well as SAE 10 and 30, and became U. S. Army Specification 2-104A. The Army now had one single type of engine oil for all ground vehicles

in three grades, instead of the two types and 13 grades with which we started. The effect of this on the supply problem is obvious, and the chances of misapplication in the field were, of course, markedly reduced. This step may well be considered a milestone in the standardization of Army lubricants.

In consideration of the low-temperature application of engine oils, it was believed that the low-temperature pumpability of the 2-104A oils should be improved to facilitate low-temperature starting and warm-up. This problem was presented to the War Advisory Committee of the Coordinating Research Council for recommendations. The result was that a project was established by the Coordinating Lubricants Research Committee to study the problem. Based on the cooperative test work that was done by this group, recommendations were made to the Ordnance Department for changes in the pour-point requirements of 2-104A as follows:

1. To lower the pour point to 0 F maximum and —10 F maximum for the SAE 30 and 10 grades respectively.
2. To include a diluted pour-point requirement of —40 F maximum for both grades when diluted with 20% ASTM precipitation naphtha.

Recommendation No. 1 was made on the basis of the fact that oils of these lower pour points were available. Recommendation No. 2 was based on the Army practice providing for dilution of SAE 30 oils at low temperatures for aircooled engines, and of SAE 10 oils at low temperatures for liquid-cooled engines. ASTM precipitation naphtha was recommended for specification purposes since it was a standard diluent available in all laboratories, and reasonable correlation was noted with gasoline-diluted blends during the CLR low-temperature pumpability tests. These recommendations were accepted and included in the specification.

The SAE 10 and SAE 30 grades of 2-104A were used in both transport and combat vehicles during the tests conducted at Camp Shilo. The results of these tests were most gratifying and indicated that the dilution pour-point requirement is a requisite for satisfactory winter operation.

A rather extensive series of dilution tests were conducted at Camp Shilo, which showed that repeated redilution of lubricants meeting U. S. Army Specification 2-104A resulted in no harmful effect on engine conditions, and at the same time assured adequate flow at the time of starting.

General field experience with the 2-104A engine oils has been very satisfactory, from operations in the desert to operations at subzero temperatures, as I have just noted. Reports are consistent as to the cleanliness of the engines at the time of overhaul, which generally is reflected in longer engine life. A constant enemy of long engine life as we all know, is dust, in which military vehicles operate a large portion of the time, and the engine life is not long in any case, as compared with commercial experience.

As noted above, U. S. Army Specification 2-104A provided for qualification of these oils prior to procurement. Such qualification followed the commercial practice where the oil supplier obtained approvals independently from the Caterpillar Tractor Co. and from the General Motors Corp., following the separate test procedures of each of these companies. The results of these tests were taken as the basis for the qualification list maintained by the Ordnance Department. It has been found desirable in the last few months to provide for qualification of these oils under direct supervision of the Army in order to conserve time, an important consideration in these days when the critical materials situation frequently means that oils must be modified and requalified quickly in order to maintain an adequate supply of these lubricants at all times. Accordingly, the specification has been modified to provide for this, and has become U. S. Army Specification 2-104B. A procedure for qualification has been prepared, and is being distributed to all oil manufacturers concerned. This procedure, or Form A, "Procedure for the Qualification of Lubricating Oils Under U. S. Army Specification 2-104B," as it is called officially, is given in Appendix I.

Briefly, the qualification procedure provides for two plans, each calling for certain engine tests. The first plan is used where the additive in the oil is unknown, and calls for all engine tests considered to be necessary for testing heavy-duty oils. The second plan may be used where the additive is known, and in this case, a limited number of engine tests are required. A known additive is one that has been fully tested previously in an oil that has successfully qualified. The engine tests may be performed in any laboratory satisfactory to the Ordnance Department, except the CRC Designation L-2-243 (Caterpillar No. 2 test), which must be run at the Armour Research Foundation. The results of all of the tests are forwarded to Armour, together with the required engine parts. Form C-1 is attached as Appendix II as a sample of the data sheets upon which the test results are reported. The Ordnance Reviewing Committee, composed of representatives from Armour, Caterpillar, and General Motors, meets at the Foundation to consider the test results and to make recommendations to the Ordnance Department as to whether the oil is satisfactory.

■ Gear Lubricants

The first grade of gear lubricant to be eliminated was the SAE 250, for which the application demand was small. After a meeting with the manufacturers of axles used in military equipment, the straight-mineral gear lubricants were eliminated, since the relative amounts used in this case were small. A single "truck-duty" type of gear lubricant would help the supply problem and obviate the possibility of the inadvertent use of straight mineral oils in hypoid gears with disastrous results. The proposed Federal Specification VV-L-761 universal-gear oils, therefore, were required for all vehicle gear applications. The exception to this was the transmission, differential, and final drives of tanks, where the SAE 50 engine oil was used. Thus the two types and seven grades of gear lubricants were reduced to one type and three grades.

It developed that the experience of several of the gear manufacturers indicated that the SAE 140 grade was not necessary, even at extremely high atmospheric temperatures. After considerable testing by the Army of the SAE 90 grade on the desert and elsewhere, the decision was made to use this for all applications where SAE 140 had been used. At first, numerous complaints were received as to excessive leakage with the SAE 90 as compared with the SAE 140 at high temperatures, but after it was learned in the field that the real offender was gear-case maintenance, particularly seals, and that a hole can't be plugged with viscosity, these complaints tapered off to practically nothing. As a matter of fact, the experience with SAE 90

has been so satisfactory, that extensive vehicle tests are now under way in the desert using the SAE 80 grade.

There became available commercially a relatively high viscosity index gear oil which met the grade requirements for SAE 80, 90, and 140 all in one product. Because of the suitability of such a material in isolated locations where a wide range of atmospheric temperatures is encountered in a short space of time, this grade of gear lubricant, SAE 80-140 as it was called, was made available for supply, and was provided for in a new draft of the proposed Federal Specification VV-L-761. Later, when the SAE 140 grade was found unnecessary, the viscosity of this oil was modified so that only the SAE 80 and 90 grades were covered by the one material, the resulting product being called SAE 80-90. This change broadened the source of the material.

As VV-L-761 was still in the proposed Federal Specification form, it was necessary to issue Ordnance Specification AXS-825 in order to facilitate the procurement of universal-gear lubricants. AXS-825 contained the same provisions as VV-L-761, and called for the SAE 80, 90, and 80-90 grades.

The cold-weather tests at Camp Shilo indicated that the SAE 80 grade of universal gear lubricant was satisfactory at all temperatures encountered when diluted. In consideration of still lower temperatures encountered in Alaska and on the Alaska Highway, it was apparent that an "arctic" grade of gear lubricant, which could be used without dilution, was necessary. This would eliminate the difficulties in maintaining, at all times, uniform dilution of the SAE 80 gear lubricant. This problem was taken up with the Coordinating Lubricants Research Committee Group on Gear Oils, and an "arctic" grade, or Grade 75 as it will be called, was recommended which could be made available commercially.

■ Greases

To meet the grease needs of Ordnance vehicles, U. S. Army Specifications 2-106, 2-107, 2-108, 2-109, and 2-110 were written for General Chassis Grease Nos. 0, 1, 2, Wheel Bearing Grease No. 3, and Water Pump Grease No. 4. These appeared to be satisfactory for their intended applications, but during the summer of 1942, the specifications for the winter greases were closely studied with respect to performance at extremely low temperatures. This problem was taken up with the CLR group on greases, and an extensive cooperative program was undertaken which included low-temperature pumpability in standard Army grease guns, tank bogie wheel bearing torque tests in the cold room, as well as various laboratory instrument tests. As a result of this excellent cooperative work, it was possible to amend U. S. Army Specification 2-106 for Grease, General Purpose No. 0 (winter chassis grease), and 2-108 for Grease, General Purpose No. 2 (wheel bearing grease) so that they would be satisfactory at extremely low temperatures.

Tests of the modified 2-106 at Camp Shilo indicated that such products were satisfactory for general chassis lubrication down to the lowest ambient temperature encountered. They were also the most satisfactory of the various lubricants tested for front-axle universal joints and tank suspension systems, provided these were first disassembled and all summer grease removed.

The modified 2-108 was found to be satisfactory for the lubrication of wheel bearings down to the lowest temperature encountered. At those temperatures below which hand packing cannot be accomplished, the modified 2-106 was satisfactory, provided a liberal quantity was placed in the wheel hub.

■ Engine Preservative Oils

Preservation of engines during shipment and storage is a problem of using oils which not only will prevent corrosion, but also will permit the engine to be operated without damage until the oil can be replaced with the prescribed grade of engine oil. A third factor which invariably creeps, but which lately has sprinted, into the problem is that of critical materials. The oil formerly used for this purpose is Oil, Lubricating, Preservative, Medium, Ordnance Department Specification No. AXS-674. Although this material is an excellent rust preventive, it is not too successful as a lubricant and further requires a rather high percentage of critical materials.

An entirely new specification was prepared to cover an engine preservative oil that would possess the dual properties of adequate lubrication and preservation without being wasteful of critical materials. This specification, designated as Oil, Engine, Preservative, Ordnance Department Specification No. AXS-934, requires qualification of the oil with respect to the supply of ingredients, lubrication qualities, and rust preventive properties. The qualification with respect to lubrication properties consists of the 36-Hr Chevrolet Test run at a crankcase oil temperature 15 deg lower than that of the corresponding grade of engine oil. The qualification as to rust prevention comprises tests run at Rock Island Arsenal to determine protection offered against corrosion in a humidity cabinet, against salt water, and against hydrobromic acid corrosion.

■ Conclusion

Several references have been made to the Coordinating Research Council, and the assistance which has been rendered to the Ordnance Department. This assistance has been of great benefit to Ordnance in solving the many varied lubrication problems which are constantly arising. The CRC Coordinating Lubricants Research Committee is helping to develop test procedures where there are none, is carrying out correlation tests to standardize and interpret existing procedures not previously standardized, is helping to lay out test programs and is assisting in the analyzing of the results of the tests. Regular meetings are held with the CRC War Advisory Committee to discuss the fuels and lubricants problems facing the Ordnance Department. This cooperative work is very effective in solving our problems.

The Ordnance Department regards the lubricants that have been standardized and their specifications in a very critical light. We must have lubricants that will perform satisfactorily under a wide variety of operating conditions. I have referred to the subzero temperature testing at Camp Shilo last winter. Observations on the Alaska Highway operations have permitted extrapolation of the data obtained to still lower temperatures. Test operations in the desert have been under way for over a year, and are continuing. These actual equipment tests in the field are supplemented by laboratory work, which may involve the complete unit or component parts carried out in a cold room, in a hot room, or in a moderate-temperature room. By maintaining this constant vigilance, we hope to keep our lubricants in fighting condition.

APPENDIX I

Procedure for the Qualification of Lubricating Oils Under U. S. Army Specification 2-104B

Form A **April 5, 1943**

THIS procedure is applicable to any oil company desiring the qualification of an engine oil under U. S. Army Specification 2-104B.

1. The oil company will request from the Technical Division, Ordnance Department, their Form B "Request for Assent to Test Engine Oil for Qualification Under U. S. Army Specification 2-104B." All expense involved in the actual running of these evaluation tests and the cost of all qualification tests regardless of where made, will be borne by the vendor of the product.

2. Form B must be executed in duplicate for each SAE grade of oil and forwarded to the Technical Division, Ordnance Department, Washington, D. C. This form covers the necessary information relative to base oil and additive required to clear the production of the finished oil with the Petroleum Administration for War.

3. If the data are acceptable, the Ordnance Department will notify the oil company of the tests required for each grade and will schedule the CRC-Designation-L-2-243 at the Armour Research Foundation.

4. The following series of tests will be required when the finished lubricating oil uses an additive not previously used in an oil approved by the Ordnance Department. Note: A product in which any change is made in the additive must be qualified as a new product.

For SAE 10 Grade:

a. CRC-Designation-L-4-243
b. CRC-Designation-L-1-243
c. CRC-Designation-L-2-243
d. CRC-Designation-L-3-243

For SAE 30 Grade:

a. CRC-Designation-L-5-243
b. CRC-Designation-L-4-243
c. CRC-Designation-L-1-243
d. CRC-Designation-L-2-243
e. CRC-Designation-L-3-243

For SAE 50 Grade:

a. CRC-Designation-L-4-243

5. The following series of tests will be required when the finished lubricating oil uses an additive which has been previously used in an oil approved by the Ordnance Department, but in a different base oil. In this case, each grade of oil must contain the additive in the same concentration as that used in the corresponding grade previously qualified. Note: A product in which any change is made in the base stock must be qualified as a new product.

For the SAE 10 Grade:

a. CRC-Designation-L-4-243
b. CRC-Designation-L-1-243
c. CRC-Designation-L-2-243

For the SAE 30 Grade:

a. CRC-Designation-L-4-243
b. CRC-Designation-L-1-243
c. CRC-Designation-L-2-243

For the SAE 50 Grade:

a. CRC-Designation-L-4-243

6. The Ordnance Department will be the sole judge of its own satisfaction as related to the tests to be required and the interpretation of the test results.

7. As a necessary first step in all cases, the requirements of the test, CRC-Designation-L-2-243 must be met. This test must be conducted at Armour Research Foundation, Chicago, Ill., and will apply to the SAE 10 and 30 grades. These tests and other necessary laboratory testing will normally require 30 gal for the SAE 10 and 30 grades, and 10 gal of the SAE 50 grade to be sent to Armour. Later instructions may modify these requirements.

8. Apart from the test, CRC-Designation-L-2-243, and the Compatability Test, which must be run at Armour Research Foundation, the tests under paragraphs 4 and 5 may be run in any laboratory approved by the Ordnance Department. (See U. S. Army Specification 2-104B, paragraph H-1.)

9. If it is the desire of the oil company to use a different concentration of additive from that present in the corresponding grade of an oil previously qualified by the Ordnance Department, the series of tests called for in paragraph 4 will be required, unless otherwise specified by the Ordnance Department.

10. The Ordnance Department, through selected advisers or representatives, will keep in touch with the progress of the tests. The Ordnance Department shall be notified of the completion dates of all tests so that they may make inspection of the test results.

11. Upon completion of all the required tests, all data and material for qualification, as covered by the "Final Test Report," Form C must be submitted to Armour Research Foundation, Chicago, Ill., for review by the Ordnance Department.

12. At the time of shipment of data and material to Armour Research Foundation, notification of shipment must be sent to the Technical Division, Ordnance Department and the Armour Research Foundation.

13. All questions as to the status of an oil submitted for qualification should be taken up directly with the Technical Division, Ordnance Department, Washington, D. C.

APPENDIX II

CRC-Designation-L-1-243

Form No. C-1 **April 5, 1943**

CLR Procedure for Determining in an Engine the Effect of Engine Oil on Ring Sticking, Wear, and Accumulation of Deposits

Laboratory Tests Questionnaire and Final Summary of Results

This questionnaire is to be filled out and attached to, and made a part of, the Final Test Report annexed hereto.

A. Fuel Characteristics

1. Brand name ....
2. Straight run, cracked, or blend ....

turn to page 368

An AIRCRAFT DOUBLE WINDSHIELD

THE problem of maintaining forward vision through aircraft windshields during flight through icing atmosphere has long confronted the operators of transport aircraft. Numerous attempts have been made to solve the problem, but none has proved adequate.

Among the means for eliminating windshield ice tried by various organizations are the following:

1. Freezing-point depressants – alcohols, mixtures of alcohol and glycerin, triethanol amine – applied to the weather-exposed surface.
2. Electrical heat – radiant and direct.
3. Heated air – directed against the inside surface of the windshield pane.
4. Windshield wipers in combination with a freezing-point depressant.
5. Low-adhesion surface.
6. Inflatable, transparent elements.
7. High-velocity air jets.

No attempt will be made in this paper to discuss all of the above. Experience to date, however, has indicated the ineffectiveness of liquid freezing-point depressants due to the sizable weight penalty that must be accepted if an adequate supply is to be provided for long flights. Also, at times when the rate of ice formation is high, antifreeze solutions must be applied copiously to prevent their becoming ineffective by dilution. The high-volatility fluids have been found to evaporate too rapidly to remain active long. In evaporating they tend to augment the freezing of moisture on portions of the windshield. The low-volatility freezing-point depressants remain on the windshield long after the icing region is passed and, as a result, are more objectionable than the ice as an obstruction to vision since the ice will melt and evaporate after descent to levels where the temperature is above freezing. The proper distribution of these fluids over the external surface of the windshield is an important factor, the accomplishment of which varies among different aircraft installations. The electrical methods of generating heat for ice prevention are considered too costly from the standpoint of weight sacrifice and of increasing the demand on aircraft electrical systems that are already heavily burdened.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 8, 1943.]

The use of heated air appears to provide the best solution to the windshield ice prevention problem at this time. This hot air may be obtained either from the airplane's heating system or from small internal-combustion heaters. The manner in which heat is extracted from the air by the windshield, however, largely influences the success of this means.

Fig. 1 illustrates an arrangement in which a flexible tube is used to direct heated air against any part of the windshield pane where ice prevention is desired. The shortcoming of this method is that it requires an excessively large quantity of hot air. Since the air impinges directly against the glass, a large portion is deflected back into the cockpit without having given up its heat for the elimination of windshield ice. As a result, the cockpit may become uncomfortably warm for the flight personnel. This method is only partially effective under moderate icing conditions and is almost without effect during severe conditions.

These objections led to the development of the double windshield.

After a review of the various means tested for preventing formation of windshield ice, it appeared that a dual-pane installation in which heated air was circulated through

VARIOUS means that have been tried to eliminate the annoyance of ice on aircraft windshields are briefly discussed by Mr. McBrien. The three chief losses that have to be compensated when heat methods are employed result from the formation of ice due to:

1. The impact of subcooled water droplets. 2. Evaporation freezing. 3. Heat lost to the air flowing over the windshield.

The development of the double windshield currently used by United Air Lines is discussed, with mention of the shortcomings of the several transparent materials used in the installation.

Experience during the past two winters has proved that adequate heat can be supplied the double windshield from the cabin heating system of the DC-3 airplane to prevent ice from obscuring vision in all except the most severe conditions encountered in scheduled operations.

The optical characteristics of the two-pane in-

—Its DEVELOPMENT and USE

by R. L. McBRIEN

United Air Lines Transport Corp.

the inter-pane space, as first proposed by the NACA, possessed the best possibilities. The reference information available in the early part of 1941 pertained only to limited flight tests with small airplanes of lower speed and of shorter range than the DC-3 airplanes used by most U. S. airlines. Accordingly, steps were taken to equip a DC-3 with a double windshield designed along the lines of that previously tested by the NACA.

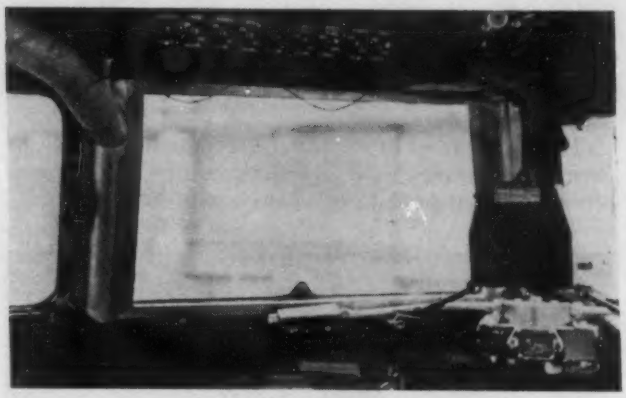
A DC-3 airplane was removed from scheduled operations for this installation and for special flight tests to follow. The fixed portion of the windshield to the left of the center-post and in front of the pilot was selected as the location for the dual-pane windshield. This section of the windshield has an area of approximately 1.6 sq ft.

The test installation is illustrated in Fig. 2. The outside pane was of ¼-in. thick Hi-Test Safety Plate Glass. The inside pane was of ⅛-in. thick safety glass. The two were installed with their surfaces parallel and with a ¼-in. air space between them. Inlet and exhaust air-headers were located at the left and right edges of the dual installation, respectively. During the first flight tests, the heated air was obtained from the cabin's heating system and was exhausted from the windshield back into the heating ducts leading into the cabin. The aim of this installation was to conserve heat by supplying to the cabin the heat remaining in the air that had not been extracted by the windshield. The dynamic pressure resulting from the flight of the airplane was utilized to maintain circulation of the hot air through the windshield and back into the heating system.

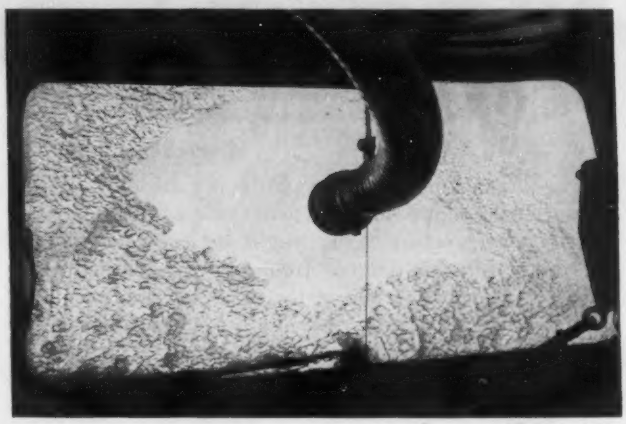
stallation are somewhat inferior to those of a single pane but are felt to be more than offset by improved ice protection and by increased strength to resist bird-strikes.

■ ■ ■

THE AUTHOR: R. L. McBRIEN is a specialist in the testing of aircraft de-icing equipment. Joining United Air Lines in 1934, he worked in the instrument department at the company's Cheyenne, Wyo., repair base, and three years later was transferred to the engineering department in the general office, Chicago, where he became project engineer. Mr. McBrien received his B. S. in mechanical engineering from the University of Oklahoma in 1933.



■ Fig. 1 – An unsatisfactory early arrangement for preventing ice formation – the flexible tube directs heated air to the part of the windshield pane which it is desired to defrost.



■ Fig. 2 – DC-3 test installation of a double windshield designed according to NACA test models

A spray of water directed onto the outside surface of the windshield was obtained by means of jets that had been installed on the fuselage forward of the windshield. By operating the aircraft at altitudes where subfreezing temperatures existed, it was easy to cause ice to form and thereby to get an indication of the effectiveness of the various combinations that were tested.

The windshield system just described, while being of some effect in preventing ice from forming, was generally disappointing. Measurements of the quantity of airflow and of the temperatures of the inlet and outlet air from the windshield indicated that a much higher rate of airflow would have to be obtained to produce the desired results.

The use of a fan or blower to force air through the system was avoided because of the weight that would be added. A survey was made of the static pressures of the outside air at several locations on the fuselage skin to locate a low-pressure region into which the outlet air from the windshield might be exhausted. A region was found that provided a pressure drop of 10 in. of water between

inlet and outlet air-headers during cruising flight. This differential produced sufficient air circulation to increase the heat supplied the windshield to the extent that ice formation was prevented at the lowest temperature (19 F) of the flight tests. The maximum amount of heat extracted by the windshield for this installation was computed as being 1852 Btu per hr per sq ft.

Other experiments were conducted with a small internal-combustion heater supplying the heated air to the double windshield. This unit made ice prevention possible with a lower rate of air circulation since it provided inlet air of higher temperature.

The effectiveness of increasing the heat supply may be seen in Fig. 3, which shows both the heated double windshield and an unheated single-pane windshield.

■ Fig. 3 – A heated double windshield (left) compared with an unheated single-pane windshield (right)

These tests appeared to justify the installation of similar windshields for use during scheduled operations into conditions where natural icing might be encountered. The service installation differed from the test installation in that the intermediate post in the left-hand windshield was removed and the glass area was made continuous from the corner post to the center V post. Heated air was introduced between the two panes at the corner post and exhausted at the right-hand edge of the left windshield. This change enabled the installation to be fitted better into the crowded cockpit. The tests indicated that an adequate supply of heat was available to provide ice prevention for a major portion of this area. As in the case of the test installation, the first service installations utilized laminated safety glass in both forward and rear panes. The first experiences in flight through natural icing conditions were favorable, in that clear vision was provided through the entire heated windshield down to the lowest reported temperature of 12 F. Other troubles soon became apparent, however. Several instances of cracked windshield panes, both outer and inner, occurred due to thermal and mechanical strains. Panes of such large areas were more critical to frame alignment than the smaller ones had been. The majority of the cases of cracked windshield panes were involved in failures of the inner pane. Because of this, it was believed that the stresses were greater in this pane due to its higher average temperature. Accordingly, several plastic panes were installed, with the thought that plastic material would better withstand the stresses imposed.

The use of plastic in this installation, however, proved unsatisfactory for the following reasons:

1. Plastic panes are not as stiff as glass panes of the same thickness. As the air leaving the double windshield is exhausted to a region of low pressure, the pressure in the inter-pane space is, under some conditions, less than cockpit pressure. This pressure differential caused an inward deflection of the plastic toward the forward pane that restricted airflow through the inter-pane space and caused uneven distribution of heat throughout the double windshield. With plastic material in the inside pane, ice prevention was much less effective.

2. The plastic inner pane possessed inherently poorer optical qualities than did the glass pane. In addition, the deflection of the plastic pane resulting from the air pressure differential existing between the cockpit and the inter-pane space presented to pilots a concave surface which, they reported, appeared to focus more annoying reflections from cockpit lights into their eyes than did the plane surface glass.

3. Probably the most objectionable feature of plastic material in a windshield installation that provides for the circulation of hot air is that the flow of hot air across the plastic surface generates on it a charge of static electricity. This electrical charge attracts dust particles carried by the air through the inter-pane space and draws them to the plastic. With the plastic inner-pane, the double windshield became, in effect, an electrostatic dust-separator, and as such was highly objected to by flight personnel because it impaired forward vision.

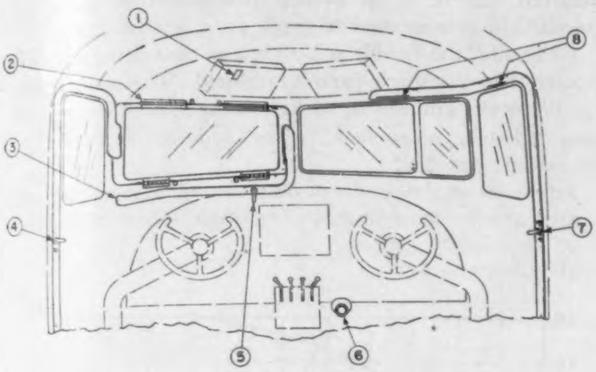
4. Plastic material becomes scratched easily and requires special care when cleaned.

With the abandonment of transparent plastic and in view of the difficulty that had been experienced in the use of conventional, annealed safety glass, it was decided to test the serviceability of heat-treated glass panes for this installation. The outside pane of the currently used double windshield is fully tempered, ¼-in. thick solid plate glass. The inside pane is of semi-tempered ⅜-in. thick laminated safety glass, composed of two ⅛-in. semi-tempered glass layers and a ⅛-in. plastic interlayer. This combination is proving satisfactory except for the relatively poor optical qualities of the laminated pane. The optical properties of this pane, however, are better than were those of the plastic pane. The comparative freedom from cracking of both inside and outside panes in this combination has facilitated installation and made the windshield quite trouble-free in service as regards the necessity of replacing cracked panes. Further, the solid plate glass used in the outside pane has proved more effective in ice prevention because of its seemingly better heat transmission than that of laminated glass.

The present assemblies of the double windshield in DC-3 airplanes operated by United Air Lines are such that the inside pane may be readily swung open by station mechanics or by pilots during flight for purposes of cleaning the two surfaces bounding the inter-pane space. This pane is held in place by hinges along both the upper and lower edges. All hinge pins are removable by hand to facilitate opening or removal of the pane. This is illustrated in Fig. 4.

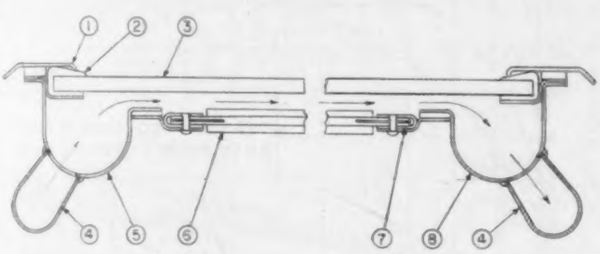
A horizontal section of the double windshield is shown in Fig. 5.

The air system of the United Air Lines double windshield is illustrated in Fig. 6. Air enters the duct in the nose of the fuselage and is conducted through the steam radiator of the cabin heating system to the windshield by



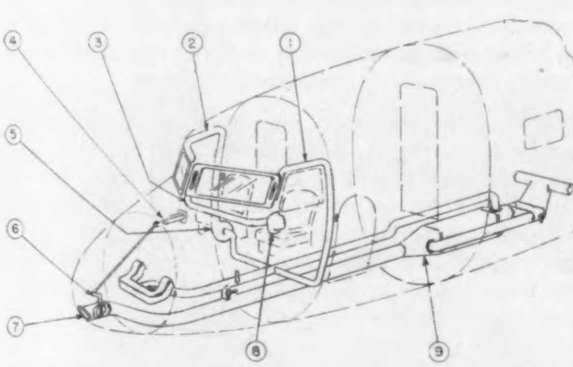
1. DEFROSTER FAN SWITCH
2. HINGE
3. AIR OUTLET
4. HOT AIR INLET VALVE
5. OUTLET VALVE
6. NOSE AIR VALVE CONTROL
7. DEFROSTER VALVE
8. DEFROSTER VENTS

■ Fig. 4 – Double windshield – cockpit airduct system



1. OUTSIDE FRAME
2. RUBBER SEAL
3. OUTER PANE $\frac{1}{4}$ INCH SOLID, FULL TEMPER
4. ADAPTER
5. HEADER HOT AIR INLET
6. INNER PANE $\frac{3}{8}$ IN. LAMINATED SEMI TEMPER
7. INNER FRAME
8. HEADER AIR EXHAUST

■ Fig. 5 – Double windshield – horizontal section



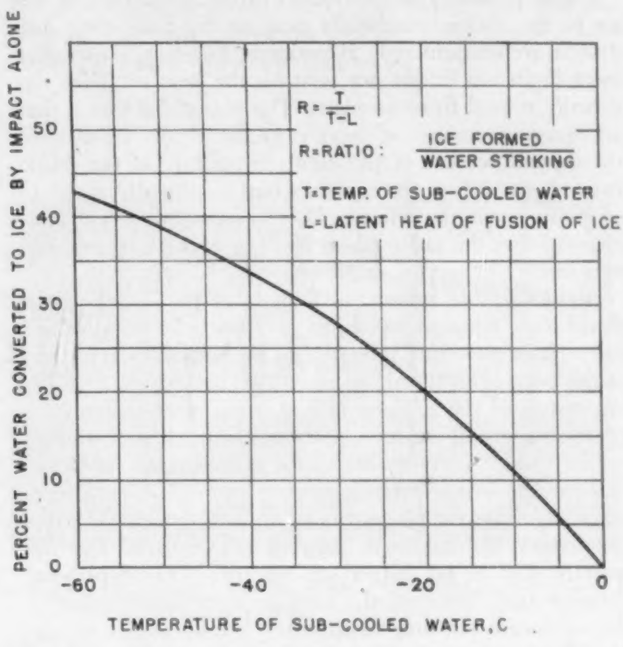
1. WINDSHIELD HOT AIR DUCT
2. DEFROSTER INLET
3. HOT AIR OUTLET
4. NOSE AIR VALVE CONTROL
5. DEFROSTER FAN
6. NOSE AIR VALVE
7. NOSE AIR SCOOP
8. EXHAUST OUTLET
9. RADIATOR

■ Fig. 6 – Double windshield – hot-air system

the ducting shown. The air control valves of the system are illustrated in Fig. 4.

Operating experience with the double windshield during the 1942-1943 winter season indicates that for most of the icing conditions encountered, adequate heat can be drawn from the DC-3 steam heating system to prevent the formation of ice over a large portion of the windshield during flight through icing atmosphere. However, there have been instances in which the rate of ice formation was greater than could be counteracted by the windshield heat. Glaze ice encountered at a rate of formation of ⅜ in. in 8 min at 12 F obscured vision during one flight and indicated that if a heated windshield is to be effective for the more severe conditions that may be encountered, more heat will have to be provided than is provided in this installation.

A consideration of the manner in which subcooled water particles are converted to ice upon impact with airplane windshields will serve to illustrate the increased demand for heat as the subcooling temperature decreases. Fig. 7 shows the proportion of the total water striking that is changed to ice by impact alone.



■ Fig. 7 – Aircraft ice formation, as influenced by the degree of subcooling

Heat loss due to evaporation freezing and the heat lost to the air stream also must be compensated. The demand for heat to prevent the formation of impact ice may be easily the greatest of the three since ice results instantaneously from impact and since the proportion of the subcooled water immediately converted to ice is greater at low temperatures.

The development of the relation shown in Fig. 7 may be of interest. The curve was calculated on the assumption that the impact of subcooled water particles with an object serves only to change their state from subcooled water to a mixture of ice at the subcooling temperature and water at freezing temperature. No heat transfer attends the impact other than that resulting from the release of the

latent heat of fusion from that part of the water converted to ice to the part that remains water. If, upon impact, a portion of the water is converted to ice, and if the heat released by the formation of this ice is absorbed in raising the temperature of the remaining water from its previous subcooled temperature to freezing, the heat exchange within a unit weight of water may be expressed as follows:

$$RL = C(1 - R)(T_f - T)$$

where:

R = Proportion of subcooled water converted to ice by impact.
L = Latent heat of fusion of ice.
C = Specific heat of water.
T_f = Freezing temperature of water.
T = Temperature to which water was subcooled.

When centigrade temperatures are used, the formula reduces to $R = T/(T - L)$; and when fahrenheit temperatures are used, to $R = (32 - T)/(L + 32 - T)$.

The very small amount of heat generated by the impact of the water droplets against the windshield has been neglected in the above considerations.

Flight personnel are instructed to direct the flow of hot air to the double windshield prior to the time icing conditions are encountered. Experience, however, shows that most flight officers do not turn on the heat until ice has actually started to accumulate. The reason for this is that an excessive amount of heat is radiated into the face of the captain because of the high temperature of the inside pane. Inasmuch as the captain's face is normally about 10 or 12 in. from the windshield, several reports have been received that the radiant heat tends to make faces hot and eyes dry.

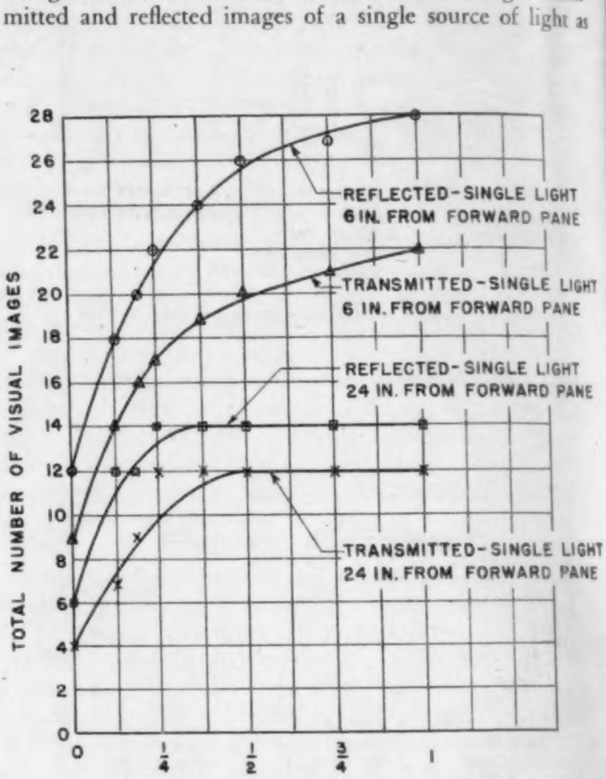
A considerable amount of fogging of the double windshield was reported by flight personnel, at first. These experiences were due primarily to the initial unfamiliarity of the pilots with the air system. Tied in with the ventilating system of the airplane as it is, some valve settings will permit warm, moist air to be drawn from either the cabin or the cockpit into the inter-pane space, where condensation on the inside surface of the outer windshield pane takes place. As the familiarity of flight personnel with this installation has increased, fogging has occurred less frequently and is no longer of concern. This experience demonstrated, however, the need for simplified valving of the air system not only to eliminate the possibility of fogging but so that minimum attention on the part of the pilot will be required.

The dual pane installation has proved considerably more resistant to bird-strikes than have the windshields with which the airplanes were previously equipped. A record of windshield bird-strikes has been maintained from October, 1942, to date. During this interval, six direct strikes have been scored. Two of these were against windshields in which were installed conventional high-test safety glass (annealed). In each instance, the bird crashed completely through the windshield, and in one case, a first officer was injured to the extent that medical attention was required. Four of the six strikes were against double windshields. In all of these instances the windshields were undamaged. The greatest airplane speed at the time of these strikes was 180 mph. The only effect of bird-strikes against the tempered glass of the dual-pane installation thus far has been obscuring of the pilot's vision because of remains smeared on the windshield.

Several comments and observations by pilots have been received relative to the optical qualities of the dual-pane windshield as compared to single-pane windshields.

Generally, the double windshield is considered optically poorer than the single-pane windshield. The main reason for this is the multiplicity of light images, both transmitted and reflected, occasioned by the addition of two more prime surfaces.

Fig. 8 shows the results of observations relating to transmitted and reflected images of a single source of light as



■ Fig. 8 – Image multiplicity resulting from the double-pane installation

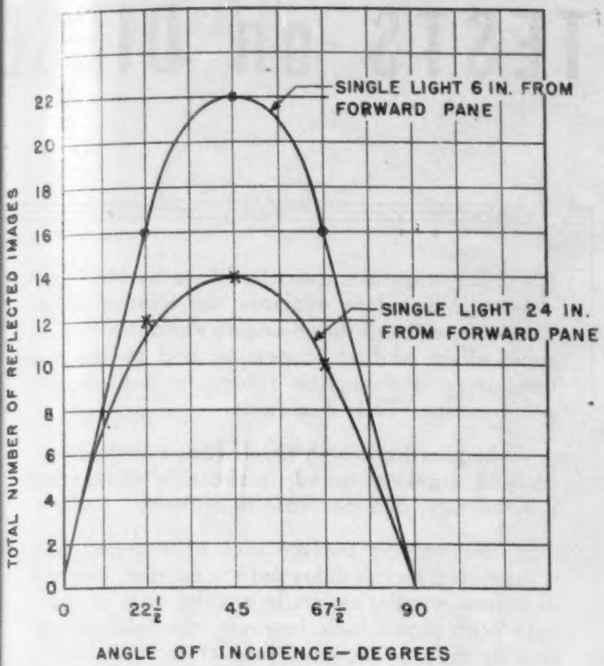
the number of images is affected by the distance of the light from the windshield and as affected by the spacing between the two panes. Fig. 9 shows the observed relationship between image multiplicity and the angle of incidence at which light strikes a dual-pane installation.

Fig. 10 is a photograph illustrating the image multiplicity that may result if the cockpit lights of relatively high intensity are located near the dual-pane installation.

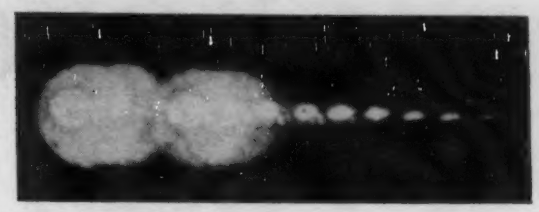
During the first use of the semi-tempered inside panes of the double windshield, reports were received from pilots indicating that a definite "prison bar" effect was observed in the windshield when polaroid goggles were worn. These "prison bars" are dark, vertical stripes spaced approximately 1.5 to 2 in. apart across the entire area of the double windshield and were found to be caused by the strains set up in the glass as a result of the tempering process.

Fig. 11 illustrates a clear pane of the windshield as normally viewed against a light background.

Fig. 12 illustrates the same pane as normally viewed against a dark background. It will be observed here that the so-called "prison bars" may be seen faintly.



■ Fig. 9 – Image multiplicity resulting from the double-pane installation (¼-in. spacing)

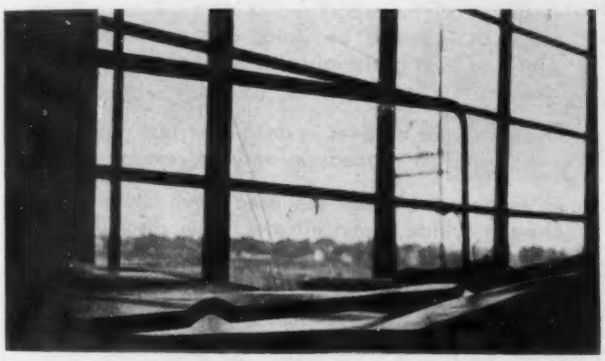


■ Fig. 10 – Image multiplicity that may result from bright cockpit lights located too close to the dual-pane installation

Fig. 13 is a photograph taken through a polaroid filter showing the "prison bars" as they appear against a light background.

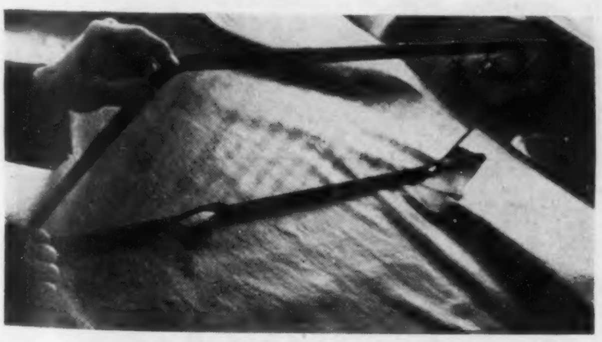
Fig. 14 is a photograph taken through a polaroid filter of the windshield pane against a dark background. This clearly illustrates the extreme case that might be encountered by flight personnel who wear polaroid goggles.

Noticeable as are the above-mentioned effects of polarization, they are not considered objectionable inasmuch as the use of polaroid goggles by flight personnel is discouraged. The increased impact resistance of tempered glass and its increased resistance to thermal and mechanical stresses are considered ample justification for its use.

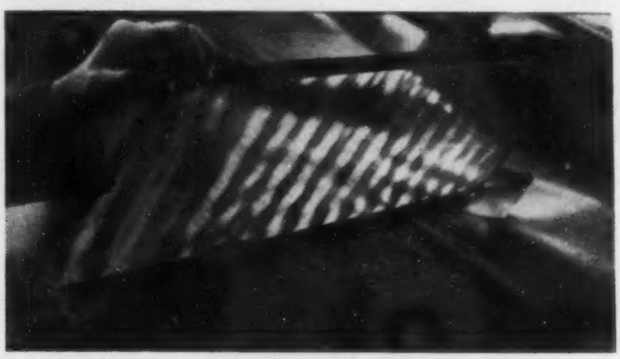


■ Fig. 11 – A clear pane of the windshield as normally viewed against a light background

■ Fig. 13 – Windshield viewed against a light background (photograph taken through a polaroid filter) – showing "prison bars"



■ Fig. 12 – A clear pane of the windshield as normally viewed against a dark background, showing faint "prison bars"



■ Fig. 14 – Windshield viewed against a dark background (photograph taken through a polaroid filter) – showing pronounced "prison bars"

COLD-STARTING TESTS on DIESE[L] [E]N[GINES]

THE investigations described here by Mr. Porter were initiated to evaluate the relative importance of pertinent diesel-engine variables, the relative effect of fuel properties, and to compare laboratory and service results on cold-starting performance. Tests disclosed:

1. Required cranking time is decreased with increased cranking speed, increase in surrounding temperature, and decrease in altitude.

2. The starting performance of undoped fuels is predicted by the delay cetane number. Increase of cetane number results in greater ease of starting. With doped fuels, however, the delay cetane number may or may not predict service starting performance.

3. It was indicated by tests on one make of engine that laboratory results may be used to predict service starting performance.

4. Various substances are effective as starting dopes, but the report indicates that an auxiliary applicator should be used, and warns that the effect of such materials on maintenance should be determined first.

The author outlines in detail the test equipment used, the test procedure, and the results.

Among the additives used were chlorine, hydrogen sulfide, amyl nitrate, ethyl disulfide, and chloropicrin. These ranged in amounts from 20 to 35 ml at ambient temperatures of 20 F, for starting in 5 sec.

★ ★ ★

THE AUTHOR: H. R. PORTER (M '37) has returned for a year to the University of California, from which he received his bachelor's degree in 1930 and his master's degree in 1932. This time he is specializing in studies on internal-combustion engines and fuels, under Prof. Carl J. Vogt. Mr. Porter joined the research and development department staff of the Standard Oil Co. of Calif. as research engineer in the engine laboratories, and has continued in that capacity.

DURING the period 1938 to 1939, our company conducted a series of investigations on the cold-starting characteristics of diesel engines and diesel fuels. These tests were initiated in order to evaluate the relative importance of the pertinent engine variables under cold-starting conditions, to investigate the relative effect of fuel properties on engine starting, and to compare laboratory and service results on cold-starting performance.

Significant fuel variables which might be considered in connection with cold-starting performance are volatility, viscosity, gravity, and ignition quality. In the following analysis, ignition quality as measured by delay cetane number is shown to be the most significant variable.

In the latter part of our investigations, a few brief tests were made on auxiliary starting means, such as adding liquid dopes or gases to the intake air. Described here are the test fuels, test equipment, and test procedures used, and also a review of the significant results obtained.

Test Fuels

The test fuels were selected to cover the following ranges of stocks, types, and physical properties:

Stocks: Eastern, Mid-Continent, and California	
Types: Straight run, cracked, and doped	
Physical properties:	
Gravity, deg API	22.1-42.8
Viscosity at 100 F	
SSU	31.8-79.7
Centipoises	0.84-14.38
Volatility at 50% point, F	330-636

Table 1 shows the pertinent inspections of the test fuels.

Test Equipment

Test Engines – Table 2 lists the general specifications of the several test engines used in conducting the tests on cold starting of diesel engines. Pictures of the laboratory test equipment are shown in Figs. 1, 2, and 3.

Starting Drive – The starting drive used on the two laboratory single-cylinder engines consisted of two Austin transmissions connected in tandem and driven by a 3 hp electric motor. Connection to the test engine was made by a flat belt on the flywheel. An overrunning clutch was built into the drive in order that the engine could run free of the starter upon firing. This arrangement was capable of cranking the engine at speeds from 60 to 600 rpm. A close-up of the starting drive is shown in Fig. 4.

The laboratory multicylinder engine and the service engines used in the tests were equipped with gasoline starting engines.

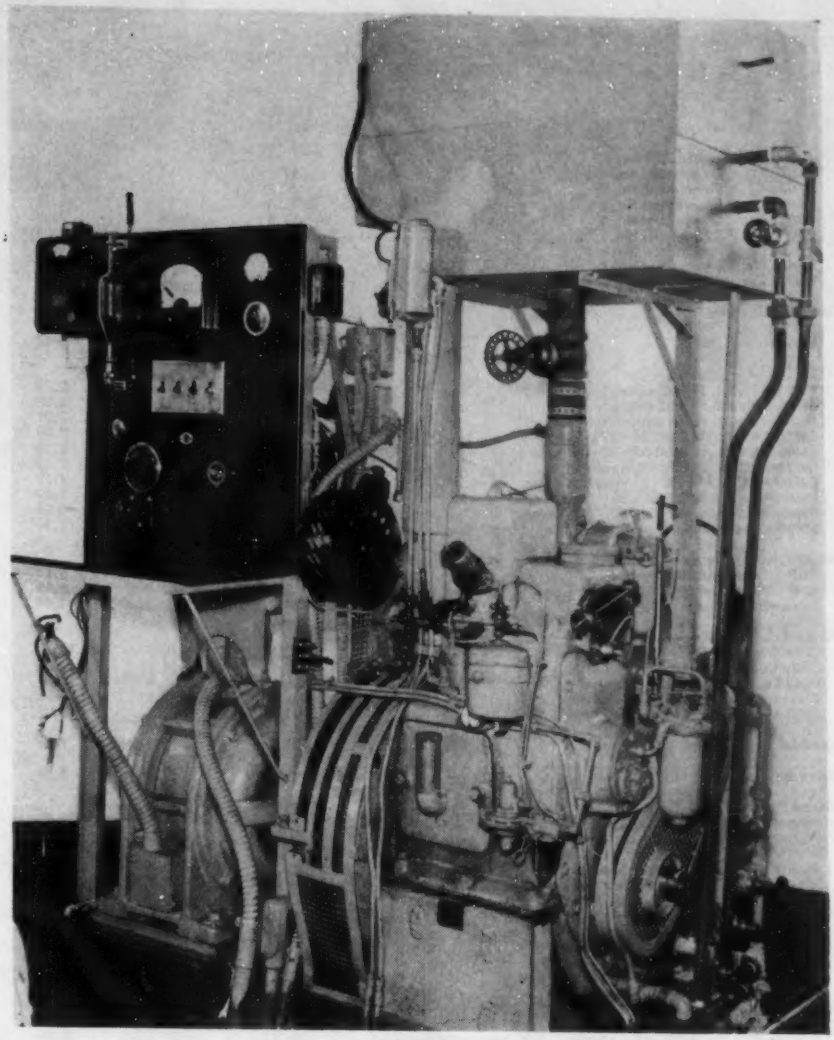
Cooling Arrangements – The temperature of the inlet air and jacket coolant were regulated by a kerosene-filled, solid-carbon-dioxide-cooled bath. The jacket coolant, which was alcohol, was circulated through a coil immersed directly in the kerosene bath. Temperature regulation was obtained by controlling the rate of flow of the coolant to

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

NGINES

by H. R. PORTER

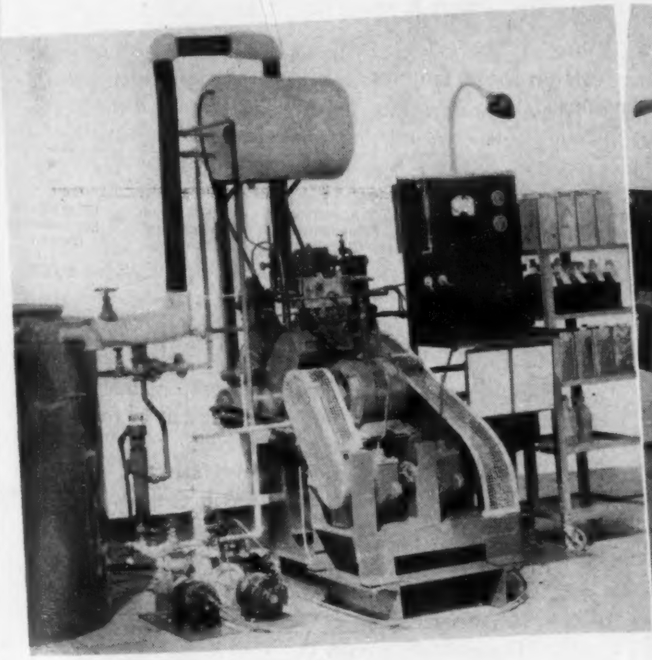
Standard Oil Co. of Calif.

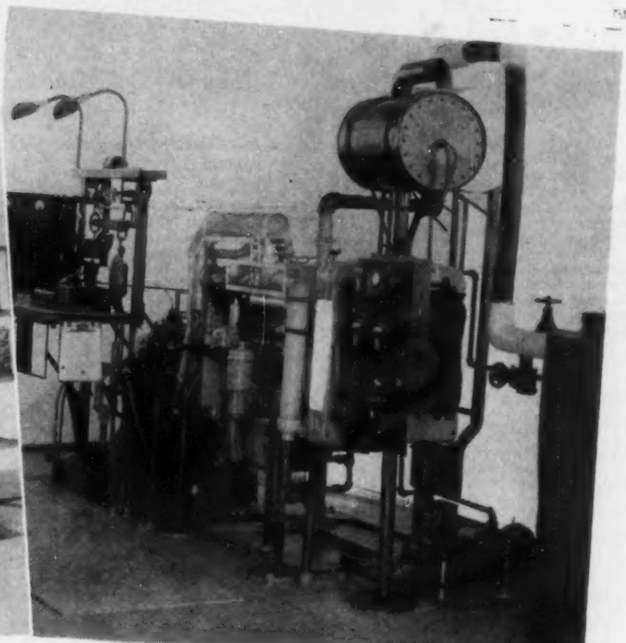


the jacket. The intake air temperature was controlled by a heat exchanger located directly over the engine and having a volume of approximately 15 gal. Coolant was circulated from the main bath to coils placed in this exchanger. The air temperatures were regulated by varying the rate of coolant flow to the exchanger. The capacity of the exchanger was such that very little increase in air temperature was obtained during the starting period. During the time required for establishing equilibrium conditions, air was forced by an electric blower through the exchanger and through the engine cylinder with exhaust valve open.

■ Test Procedure

Single-Cylinder Laboratory Engines – The general operating procedure





■ Figs. 1, 2, and 3 – Close-up (above) and two views of some of the laboratory test equipment

Table 1—Inspections of Test Fuels

	Gravity, deg API	Viscosity at 100 F: SSU	Viscosity at 100 F: Centipoises	ASTM Distillation: 10%	ASTM Distillation: 50%	ASTM Distillation: 90%	Delay Cetane No.
Pacific Coast Commercial Fuels							
Pacific Coast Commercial No. 1	34	37.6	2.78	459	514	620	46
Pacific Coast Commercial No. 2	29.7	40.1	3.63	453	547	663	46
Pacific Coast Commercial No. 3	31.4	39.6	3.47	472	528	636	44
Pacific Coast Commercial No. 4	31.2	40.1	3.59	461	544	657	45
Pacific Coast Commercial No. 5	30.1	41.2	3.94	433	564	646	42
Pacific Coast Commercial No. 6	29.4	42.8	4.42	494	560	660	41
Pacific Coast Commercial No. 7	33.8	34.9	1.94	396	466	580	38.5
Pacific Coast Commercial No. 8	31.2	37.9	2.92	436	518	627	41.5
Western Gas Oils							
Western Gas Oil "A"	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil "B"	35.4	36.6	2.46	436	496	640	53.5
1936 Exchange Samples							
Mid-Continent, 35 - 37 viscosity	38.7	36.1	2.25	432	523	595	50
Mid-Continent, 50 - 52 viscosity	32.7	49.6	4.02	524	628	697	50
Mid-Continent + 1% ethyl nitrate	33.2	35.1	2.01	441	493	560	58.5
California Straight Run, 35 - 37 viscosity	35.9	36.4	2.48	432	500	616	52
California Straight Run, 50 - 52 viscosity	28.2	50.2	6.50	494	602	726	45
California Straight Run, high volatility	39	33.7	1.49	403	448	498	46
California Cracked, 35 - 37 viscosity	30.6	35.3	2.12	438	471	562	34
Pennsylvania Straight Run, 35 - 37 viscosity	38.3	41.7	3.89	512	581	654	65
Pennsylvania Straight Run, 50 - 52 viscosity	35.7	51.4	6.54	568	634	686	64.5
68% Secondary Reference in α — β methyl naphthalene	29.4	34.8	1.96	445	480	600	52
High-Volatility Stocks							
10-P Thinner Sample 1	42	31.8	0.85	320	330	355	28
10-P Thinner Sample 2	42.8	31.8	0.84	320	330	348	31.5
Commercial Stove Oil	36.2	33.5	1.46	377	437	506	36.5
Related Cuts							
Close Cut No. 1 from Type "A" Crude	22.1	79.7	14.38	607	636	680	38
Close Cut No. 2 from Type "A" Crude	33.3	34.0	1.84	413	440	492	33
Long Cut from Type "A" Crude	26.3	43.5	4.71	465	579	641	34
Close Cut No. 1 from Type "B" Crude	40.8	33.0	1.25	372	418	485	43
Close Cut No. 2 from Type "B" Crude	34.2	39.5	3.35	515	544	597	52.5
Cracked Side Cut	30.5	35.8	2.56	456	480	561	33
Secondary Reference Fuel	42.1	37	2.66	446	525	620	75
Doped Fuels							
Western Gas Oil (35.5 cetane)	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 0.1% chloropicrin	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 0.5% chloropicrin	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 1.0% chloropicrin	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 2.0% chloropicrin	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 0.5% ethyl nitrate	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 1.0% ethyl nitrate	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 2.0% ethyl nitrate	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil + 2.5% ethyl nitrate	29.3	38.1	3.03	445	514	635	35.5
Western Gas Oil (40 cetane)	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 2% chloropicrin	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 2% of a disulfide	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 2.5% sulfur	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 1% ethyl nitrate	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 1% ethyl nitrate + 1/2% bromocyclohexane	31.3	37.8	2.90	434	516	627	40
Western Gas Oil + 1% amyl nitrate	31.3	37.8	2.90	434	516	627	40

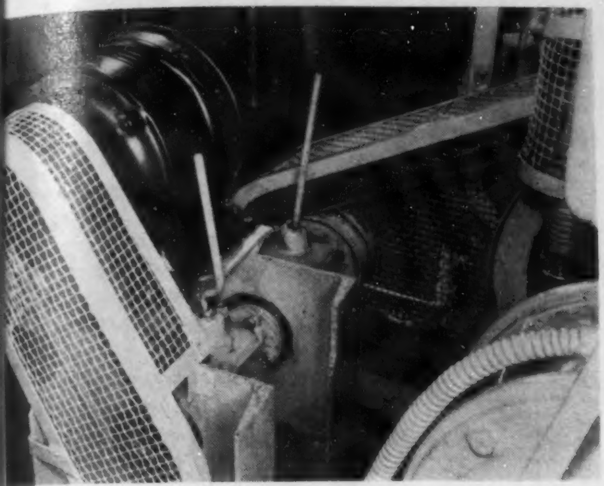
Table 2—Cold-Starting Tests on Diesel Engines
Test Engines

	Laboratory Engines			Service Engines			
Engine No.	1	2	3	1	2	3	4
Number of Cylinders	1	1	4	4	3	6	6
Horsepower	10 at 1200 rpm	8 at 1400 rpm	41 at 1400 rpm	41 at 1400 rpm	51 at 850 rpm	108 at 850 rpm	108 at 850 rpm
Combustion-Chamber Type	Turbulent Prechamber	Precombustion Chamber	Precombustion Chamber	Precombustion Chamber	Precombustion Chamber	Precombustion Chamber	Precombustion Chamber
Compression Ratio	16.8:1	16:1	16:1	16.0:1	15.5:1	15.5:1	15.5:1
Bore and Stroke, in.	4 1/4 x 6	4 1/4 x 5 1/2	4 1/4 x 5 1/2	4 1/4 x 5 1/2	5 3/4 x 8	5 3/4 x 8	5 3/4 x 8
Injection Advance (BTC), deg	25	23	23				
Injection Pressure, psi	1700	1500	1500	1500	1750	1750	1750
Starting Means	Hand		Gas Engine	Gas Engine	Gas Engine	Gas Engine	Gas Engine

employed in making the cold-starting tests in the single-cylinder laboratory engines was as follows: The jacket and inlet air temperatures were established with the engine stopped, by circulating the coolant through the cylinder jacket and by blowing cooled air through the cylinders; the engine was then cranked at the test speed for three revolutions with the compression release open to establish airflow; the compression release was then disengaged and the throttle opened. The interval between the first injection, as evidenced by nozzle chatter, and the first firing explosion was measured and recorded as the cranking time. All tests were made at full throttle and with the injection advance at the manufacturer's recommended setting. Ample time was allowed for establishment of the new test conditions before a start was attempted.

Following are the test conditions selected for investigating the effect of fuel properties in the two laboratory single-cylinder engines:

Engine No. 1	
Cranking speed, rpm	109
Cranking time, sec	10
Temperature range, F	20 to 125
Engine No. 2	
Cranking speed, rpm	280



■ Fig. 4 – Close-up of the starting drive

Cranking time, sec	5
Temperature range, F	20 to 80

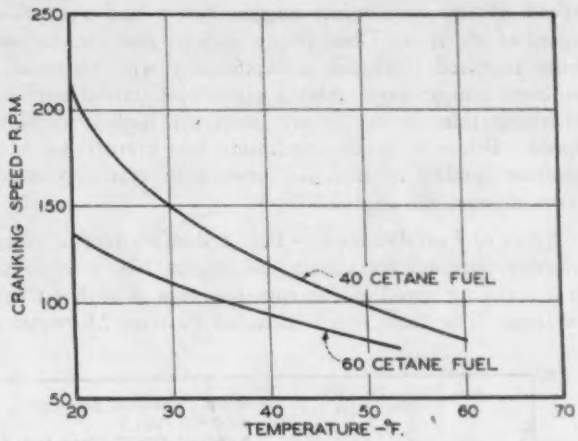
Multicylinder Laboratory Engine – The procedure employed in making cold-starting tests in the laboratory multicylinder engine No. 3 was as follows: The jacket and inlet air temperatures were established with the engine stopped by circulating coolant through the cylinder jacket and by forcing cooled air through the intake manifold; the starting engine was warmed up for 30 sec, then engaged and allowed to crank the diesel engine until it started. Attempts were made to start the diesel engine by opening its throttle for 10 sec at each 5 F increment of increase of jacket temperature. The cranking time required was considered as the time elapsed from first engaging of the starting engine until the diesel engine fired and continued to run under its own power. This was a continuous test with gradual temperature increase on each fuel, since the coolant jacket of the starting engine was connected with that of the diesel engine.

Service Engines – Starting tests were conducted on four test fuels covering a range from 40 to 75 cetane in each of four tractors at the minimum temperatures prevailing near Klamath Falls, Ore., and Alturas, Calif., during March, 1939. The locations of the tests were at an altitude of 5000 ft. Prior to shutting down at night, each tractor was operated on the test fuel until the fuel system was entirely filled. The following morning, after storing the tractor outside, the gasoline starting engine was started and immediately engaged with the diesel engine. After 30 sec of cranking, the throttle was opened fully for a period of 10 sec. If the engine did not start, injection was stopped and cranking was continued for another minute, at which time starting was again attempted. This procedure was repeated until starting was obtained. The cranking speed was measured during each 10-sec period with a calibrated tachometer and the jacket temperature was recorded each minute.

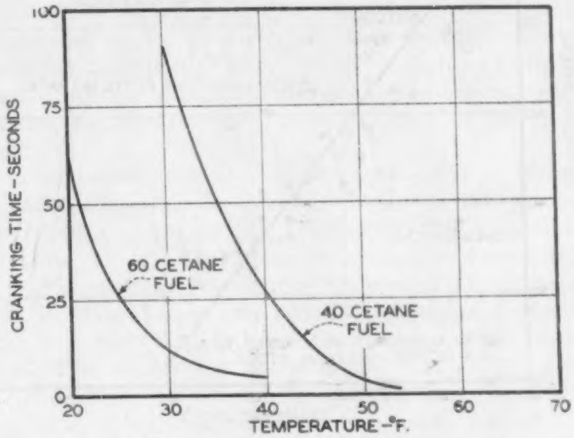
■ Test Results

Relative Effect of Engine Variables – Figs. 5, 6, and 7 show results of tests in laboratory engines Nos. 1 and 2 on the relative effect of engine variables. These include ambient temperature, cranking time, and cranking speed.

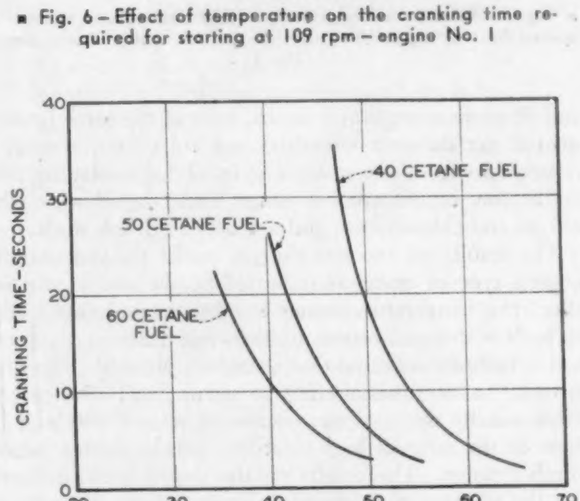
Fig. 5 shows plots of the effect of ambient temperature on the cranking speed required for starting of laboratory engine No. 1 in 10 sec. These results indicate that the cranking speed required decreased with increase in ambient temperature for starting in a given time. These results also indicate for the normal range of temperatures that for any fuel there was a minimum speed below which the



■ Fig. 5 – Effect of temperature on the cranking speed required for starting in 10 sec – engine No. 1



■ Fig. 6 – Effect of temperature on the cranking time required for starting at 109 rpm – engine No. 1

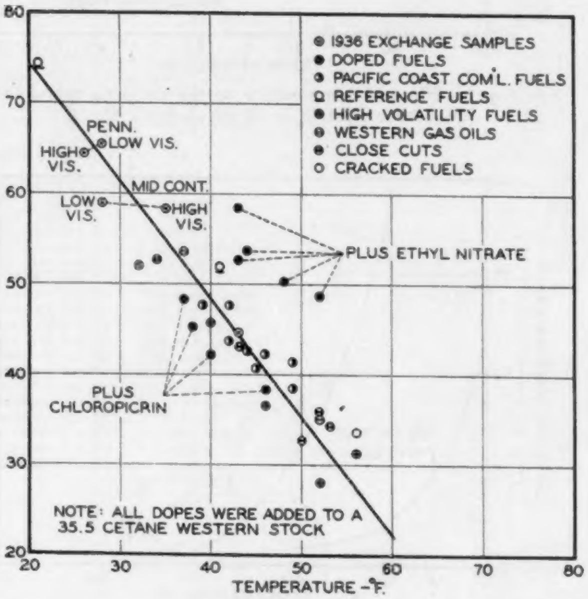


■ Fig. 7 – Effect of temperature on the cranking time required for starting at 280 rpm – engine No. 2

engine could not be started with 10 sec of cranking. Differences in ease of starting increased with decreases in ambient temperature. In the range from 30 to 60 F with a 40 cetane fuel, a decrease of 1 F would be offset by an increase of 3 rpm in cranking speed.

Figs. 6 and 7 show plots of results obtained in laboratory engines Nos. 1 and 2 on the effect of ambient temperature on cranking time required. Engine No. 1 had a cranking speed of 109 rpm, while engine No. 2 had a cranking speed of 280 rpm. These results indicate that the cranking time required increased asymptotically with decrease in ambient temperature. Also a given fuel started with less cranking time in the engine with the highest cranking speed. Below a given minimum temperature for each ignition quality of fuel, no amount of cranking would serve to start the engine.

Effect of Fuel Properties – Fig. 8 shows a plot of cetane number required for starting of engine No. 1 in 10 sec at a cranking speed of 109 rpm in terms of ambient temperature. The fuels tested included Eastern, Midwestern,

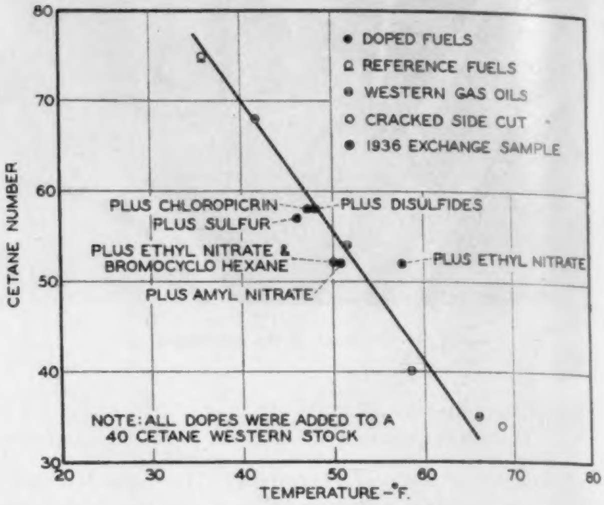


■ Fig. 8 – Relation between temperature and cetane number required for starting in 10 sec at 109 rpm for various fuels – engine No. 1

and Western straight-run stocks, fuels of the same ignition quality but different viscosities and volatilities, a series of related cuts from one stock, a series of high-volatility fuels in the low cetane number range, fuels doped with ethyl nitrate and chloropicrin, and a cracked side-cut stock.

The results on the straight-run stocks showed that the general type of stock, as indicated by the source, did not affect the temperature-cetane number relationship. Tests on fuels with equal cetane numbers but different viscosities and volatilities indicated that these two physical properties, as such, had no unusual effect on starting. This fact is also borne out by the tests on a series of related cuts and by those on the series of high-volatility fuels in the low cetane number range. The results on the doped fuels indicated that the addition of chloropicrin improved the starting performance in line with the increase in delay cetane number. Although ethyl nitrate improved the cetane number, it did not improve the starting performance.

Fig. 9 shows a plot of results obtained in engine No. 2 on the relation between ambient temperature and cetane number required for starting in 5 sec at a cranking speed of 280 rpm. The test fuels included a series of Western gas oils, a cracked side cut, a secondary reference fuel, and

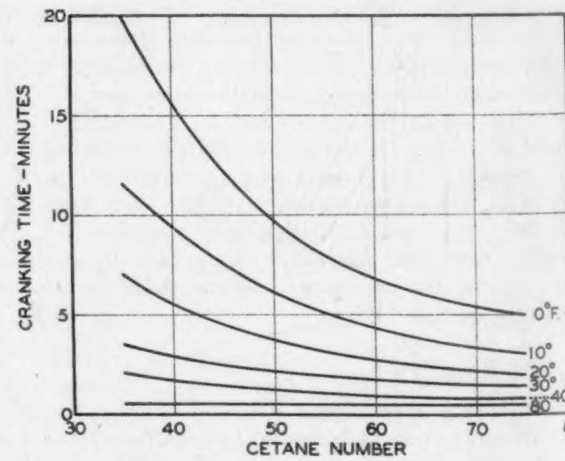


■ Fig. 9 – Relation between temperature and cetane number required for starting in 5 sec at 280 rpm for various fuels – engine No. 2

fuels doped with chloropicrin, amyl nitrate, sulfur, a disulfide, ethyl nitrate, and ethyl nitrate along with bromocyclohexane.

The results show that the starting performance of Western gas oils throughout a rather wide ignition quality range were in line with their cetane numbers. The addition of chloropicrin, amyl nitrate, sulfur, and a disulfide improved the starting performance as predicted by their corresponding cetane number improvement. Ethyl nitrate in this engine, as in engine No. 1, did not improve the starting performance although it did increase the cetane rating. However, the addition of bromocyclohexane, which in itself did not affect the starting performance, caused a fuel doped with ethyl nitrate to have a starting performance in line with its cetane number.

Fig. 10 shows plots of the results of starting tests made in laboratory engine No. 3. These results, which were



■ Fig. 10 – Effect of cetane number and temperature on required starting time – engine No. 3

obtained on four fuels covering a cetane range from 35 to 75, show cranking time in terms of delay cetane number for a range of temperatures from 0 to 80 F, and a cranking speed of 280 rpm.

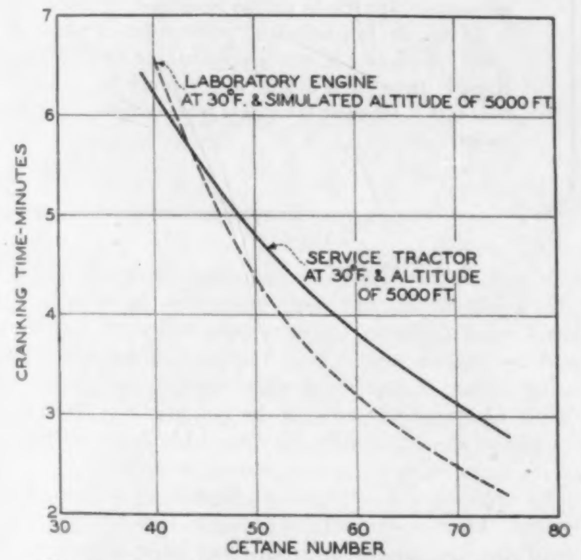
These tests show that for a given temperature the cranking time increased progressively with decrease in cetane number. The starting times required for a given cetane number were also progressively greater with decrease in ambient temperature.

Tests were also made in this engine to determine the cetane number-cranking time relationship at a simulated altitude of 5000 ft. This was obtained by throttling the intake air. A summary of these results is listed in the following tabulation, which shows average values of the differences in cranking time for the cetane ranges indicated over a temperature range from 10 to 40 F.

Cetane Range, Cetane No.	Cranking Time at 5000 Ft Minus Cranking Time at Sea Level, min (average)
75	1
55	1¾
35	4¼

From this tabulation, it may be noted that increase in altitude increased the cranking time required with lower cetane fuels to a greater extent than that required with higher cetane fuels.

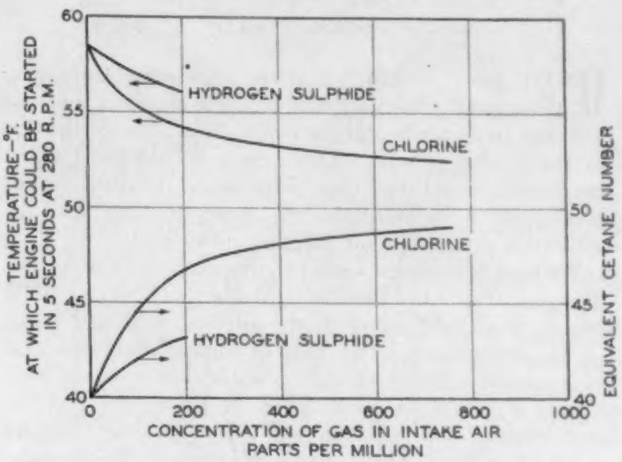
Comparison of Laboratory and Service Tests – Fig. 11 shows a plot of the average results of field tests made on



■ **Fig. 11 – Comparison of laboratory and service tests on effect of cetane number on cranking time required**

four fuels covering a range of 40 to 75 cetane numbers in four tractors at an altitude of 5000 ft and at a prevailing temperature of approximately 30 F. Also shown is a plot of the laboratory results obtained in laboratory engine No. 3 at a simulated altitude of 5000 ft and at 30 F. These results indicate for this one set of conditions, at least, that the laboratory results were in good agreement with those obtained in the field.

Additives to Intake Air – Fig. 12 shows the effects of additions of chlorine and hydrogen sulfide in the intake air on the starting performance of engine No. 2 when injecting a Western gas oil. Also shown are the equivalent cetane numbers for the various additions of the gases, as obtained from the relationship between ambient temperature and cetane number shown in Fig. 9.



■ **Fig. 12 – Effect of addition of chlorine and hydrogen sulfide in intake air on starting performance – engine No. 2**

It will be noted from these results that addition of chlorine up to 800 parts of chlorine in 1,000,000 parts of intake air was effective in improving the starting performance. Hydrogen sulfide was also effective in improving the starting performance but not to as great an extent as chlorine.

Investigations were also made on the effect of various liquid additives in the inlet air stream on the starting performance of engine No. 2 when injecting a 40-cetane Western gas oil. These tests were made at an ambient temperature of 20 F and at a 280 rpm cranking speed. The liquid additives were added by means of a burette that discharged directly into a short section of the intake pipe. This section was loosely packed with copper wool. Cranking time was measured from the moment the liquid dope was added until the engine first fired. The relative effectiveness of the various dopes was evaluated by determining the quantity which was required to accomplish starting with 5 sec of cranking.

The following tabulation summarizes test results:

Dope Added	Ambient Temperature, F	Amount of Dope Required for Starting in 5 sec, ml
Ethyl Disulfide	20	35
Amyl Nitrate	20	20
Chloropicrin	20	20

From an extrapolation of the cetane number-ambient temperature relationship shown on Fig. 9, it was found that a 100 cetane fuel would just start at 20 F with 5 sec of cranking. On this basis, the improvement obtained by the introduction of the dopes in the intake pipe was equivalent in each case to a cetane increase of approximately 60 numbers.

■ Conclusions

1. The required cranking time is decreased with increase

turn to page 368

The ECONOMICS of POST-WAR C

LOWER costs on transport of higher classes of surface freight and express through better planned equipment will be of paramount importance in the post-war period, Mr. Sheehan declares. Airline replies from a recent survey, he says, show that 83% of the carriers favor cargo-carrying facilities in passenger planes for the post-war period, particularly for transcontinental and transoceanic operation, and even for feeder line service. For the lowest ton-mile costs, however, exclusive cargo planes are the ultimate goal. Larger combination planes, for the time being, will doubtless prove most practical during the initial stages of domestic cargo development.

Curves and charts showing the effect on ton-mile costs of different size planes with varying power units emphasize the principle that the largest airplane operates at the lowest ton-mile cost. But he stresses the point that speed is another governing factor in cargo haulage. Where speed is of major importance, twin-engined planes are most efficient. A governing factor on four-engine ships is take-off weight, which must be limited, so that CAR landing weight is not exceeded for short ranges.

UNTIL peace returns, we in the aviation industry have gladly and wholeheartedly pledged ourselves and our activities to the early and successful prosecution of the war by the United Nations. This is as it should be. Until it has been accomplished there shall be no deviation in our efforts, no sidetracking of the issue; nothing shall be allowed to deter us in our progress to this goal.

We have been given a unique, though unwanted, opportunity to prove (and there is no doubt that it is constantly being proved) the worth of the airplane as a vital factor of war strength. It is not only in combat operations that its importance has been demonstrated but it is, as well, playing a part of no small degree in solving the problem of distribution of war materials to the far-flung theaters of activity. Many of these vital supplies are being transported to the fighting outposts of every corner of the world in airplanes made available through the backlog of the domestic commercial air transport system. Although many of these ships were never intended for cargo purposes, the necessity of over-night conversion and improvising with what was on hand has resulted in broadening our experience and our knowledge of the problems of the various phases of the transportation of cargo – this, not only for immediate military uses but for the commercial post-war era as well.

We have gained this knowledge with, may we say, an utter disregard of costs, for cost has given way to the necessity of delivery – delivery of vital replacement parts for tactical aircraft, tanks, and other implements of warfare, as well as goods and supplies indispensable to our fighting forces. The Army Air Forces have become the proving ground for many phases of tomorrow's commercial air cargo services. As one Army man recently put it: "You boys are getting quite a break, pioneering the cargo transport field at the taxpayer's expense." That's true, of course, but it is also true that it is to the eventual benefit of this same taxpayer.

Post-war prospects present the aspect of another challenging opportunity, for there are many problems confronting us in the reconstruction period in which distribution again will play an important role. It may even be that in this coming period the visionary possibilities of air cargo transport will become tangible realities. Thus, it is not too early to study the conditions which are likely to prevail after the war.

In considering the part the airplane will play in world and domestic commerce tomorrow, the present attitude toward the cost item will be reversed – this item will then be of paramount importance. But before we get into the analysis of the economic aspect of post-war carriage of air cargo, let us consider what amounts and kinds of cargo we foresee as the airplane's potential as a cargo carrier in times of peace, and how they will be affected by the necessity of speed in this era.

[This paper was presented at the SAE Air Cargo Engineering Meeting, Chicago, Ill., Dec. 8, 1942.]

It was pointed out in an air cargo study recently completed that the strategic cargo required to be exported from the United States in the next year will total between 40 and 45 million tons. This reaches a total movement – using airline distances – of approximately 220 billion ton-miles. Of this, about 20% of the non-military commodities for export was suitable for air cargo. However, about 50% of the present-day military exports is so suitable.

In 1941, Class I railroads originated 1.2 billion tons of freight. Of this amount, 684,000,000 tons (56%) constituted coal, ore, and other products of mines which, on cost basis, would not be shipped by air.

An additional 189,000,000 tons, or 15%, consisted of products of agriculture, animals, and forest products, which also to a large degree, will not be subject to air shipment in the near future. From this latter classification, perishables – fruits, meats, and so forth – will be eligible for air shipment, and foods generally, particularly if strides are made in dehydration.

The bulk of rail freight which would be subject to air freight competition would be in the field of less-than-carload lots, and manufactures and miscellaneous items. In 1941, l.c.l. freight totaled over 18,000,000 tons. With manufactures and miscellaneous items amounting to more than 336,000,000 tons, a gross total of 355,000,000 tons (or

CARRIAGE OF AIR CARGO

by J. V. SHEEHAN
Lockheed Aircraft Corp.

The twin-engine plane has its best advantage under 250 miles, and beyond that figure four-engined plane efficiency improves. At ranges of 500 miles, he shows that, on a four-engine plane, a 20-mile difference in speed affected operating costs about 5¢ a ton-mile, and on a two-engine plane, the same speed differential caused a cost differential of about 2¢ per ton-mile. Twin-engine planes are the most economical if size of payload does not exceed capacity of largest twin power-plants available.

■ ■ ■

THE AUTHOR: J. V. SHEEHAN has been in the aviation field since 1928, the year in which he started in the aviation division of the Ford Motor Co. Since then he has been connected successively with Universal Fliers, Ludington Airlines and the Boston & Maine Airways. In the middle of 1938 he joined Lockheed as acting foreign sales manager, and in 1939 he organized Lockheed's Market Research Department – forerunner of the Industrial Research Division which he now heads. Much of Mr. Sheehan's time now is taken up in studies which better fit the Lockheed organization to fulfill requirements of Army air transport.

the equivalent of less than 30% of total rail tonnage originated) moved over the Class I roads.

It is our belief that the airplane potential in the over-all transportation scheme is of a supplemental nature and not one to replace the railroads or the steamships. The airplane's claim to carry its part of the immense total of tonnage is too sound to be jeopardized by wild claims that it is ready to haul heavy commodities – such as products of the forests and mines – in times of peace, across the face of the globe.

It is expected that certain refinements of our present engine, propeller, and aerodynamic designs will be developed for more efficient airplane performance, thereby lessening operating costs. These factors, however, are limited, and it would not be safe to assume that they would bring the operating costs of aircraft down to compete with the lower classes of materials. Let us consider, for example, the l.c.l. freight field which averages about 4¢ a ton-mile. As I have mentioned, this traffic amounted to around 18,000,000 tons carried on the Class I railroads in 1941. While this is 1½% of the total freight carried, it represents about 2000 times as much as the total air express carried by all domestic airlines last year. If we were *able to compete on an economic basis,* and suppose we had an airplane with a payload of 16 tons, which cruised at 250 mph, and assuming that we could fly each plane 3650 hr per year, it would take 620 of these planes to move the 18,000,000 tons of l.c.l. freight. This, of course, does not take into account the rail express or mail traffic. Therefore, we should concern ourselves at present with the planning of equipment on which the cost of operation would permit us to compete for these higher classes of surface freight, express, and mail markets.

An analysis of a market survey of the domestic operators which we recently made indicates that 83% of the carriers believe cargo-carrying facilities should be considered a part of future passenger airplane design for domestic trunk line and primary connecting operations in the post-war period. But, as previously pointed out, the war has forced us to extend the scope of our markets and has advanced our research and practical knowledge by some 10 years. Consequently, many of these operators have expanded their conception of future activities to include transoceanic operations; and a number of carriers have added that the process of rehabilitation and reconstruction in foreign countries and the re-establishment of those markets after the war would require a special cargo airplane, in view of the necessity of speed in the delivery of essential foods – particularly perishable foods – and goods in great quantities suitable for airplane transportation. Although the surface competition could render a service at a lesser cost, the speed element in the transport of such commodities would justify the premium involved by such operation.

Further analysis of this survey showed that, as well, a combination passenger-and-cargo plane for the so-called "off-line" or "feeder-line" service was wanted. From both the operators' and manufacturers' viewpoint, this would seem to be a logical step in the development of this service, and it is possible that this plane could be used also for transportation inland in the isolated or normally inaccessible areas in foreign countries.

As we have pointed out, the operators' desire – lacking any experience index in the field of solely air cargo transportation – is for a combination passenger-cargo plane for domestic service. Although we can understand their reasoning in arriving at this conclusion, from studies which we have made, we are not entirely in agreement with this thinking for a combination-type airplane in certain weight classifications.

We believe that in order to attain the greatest efficiency – that is, the lowest ton-mile cost – we should use the sharpest tool to accomplish the particular job. And, in this case, we believe that ultimately the sharpest tool must be the exclusively cargo plane. Although we agree that a combination passenger-and-cargo plane would have many of the features which go to make up either low seat-mile cost or low ton-mile cost, we nevertheless believe that the basic advantages to be gained in efficient cargo operation can best be obtained in the preliminary design of an exclusively cargo plane.

Many of the essential factors, such as stowage and handling facilities and their supplementary features, which go to serve for the efficient operation of cargo ships, may not serve as the sharpest tool for the efficient operation of passenger airplanes, and vice versa: For example, the design necessary for increased payload and stowage, for additional hatches – all of which serve to lessen speed – may not be practical for passenger operation. Carrying this further, a great deal depends upon the nature of the routes: whether they are primarily large potential passenger or cargo territories. It may be that the peculiar requirements of the passengers, on the one hand, and those of the cargo, on the other, may differ greatly. For instance, it may be that the greater volume of passenger traffic desires to depart and arrive at its destination at hours most suitable to passenger requirements – that is from 5 p. m. to 9 p. m. – whereas it may be more advantageous for cargo to be picked up and delivered at the airport and depart at a later time, so long as it fits in with an early morning delivery, which might be any time between 12 and 4 in the morning.

In defense, then, of the operators' desire for an airplane which they feel most suited to the immediate post-war needs, from the standpoint of size, our studies showed that, in general, the greater the size developed efficiently around any given horsepower, the lower the unit operating cost would be. Their reasoning for a large combination airplane is to accommodate the increased passenger business and, at the same time, get their feet wet, so to speak, in the cargo field. They do not expect that there will be a sufficient volume of business in the latter field at first – that is, not until the manifold problems of regular domestic cargo operations have been solved – to warrant the purchase of a separate cargo airplane.

The problem is not only in refining the various aspects of the airplane to lessen transportation costs in order to compete with express and cargo markets but there are other indirect factors as well. A few of these problems, the solving of which would result in a lowering of rates, are a nation-wide pickup and delivery service, a more uniform method of stowage and transshipments, adequate handling facilities, and the education of the public to the habit of shipping by air. They believe that a larger combination plane providing greater passenger seating capacity and, as well, cargo, would give the greatest efficiency and lower unit operating cost during the initial stages of domestic cargo development. It is on this theory, then, that the following studies have been developed.

Our surveys indicate that the cargo field will comprise four categories: 1. Transoceanic; 2. Transcontinental; 3. Primary feeder; and 4. Secondary feeder. Since our survey dealt with the aspect of the domestic market – and as the ramifications of transoceanic operations are many – because of the limitation of space and for illustrative purposes, we shall deal only with the domestic categories.

In examining the airplanes best suited to accomplish these tasks, we analyze them from various aspects, using low ton-mile cost as the criterion: first, from the standpoint of size; second, number of engines; third, payload; fourth, range; and fifth, speed. Incidentally, in reviewing all of these assumed planes, it should be pointed out that they can be licensed for passenger transportation in accordance with the present Civil Air Regulations. The reference to cargo airplanes in the following studies should be interpreted to mean ones which would carry passengers and cargo.

The criterion used in this paper for measuring design variation is operating cost per ton-mile of payload. Payload figures were based upon airplane operation at 50% normal rated power, with zero wind and zero fuel reserve. The operating costs were computed by means of a set of cost-estimating formulas originally developed for computing and forecasting cost of passenger airplane service but modified here for application to cargo operation. Owing to the scarcity of data available on air cargo services, there are a number of unknowns in the formulas, hence, as an absolute measure, their accuracy may be somewhat lessened; however, it is believed that the formulas nevertheless provide a valid comparison of the various airplanes studied. The costs charted include all those described as direct flying costs in the former system of accounts used by the airlines. These are direct flying operations, direct flying maintenance, and direct flying depreciation expense. In addition, the total operating cost figures used here include an allowance for overhead expenses, embracing indirect operations, maintenance, and depreciation, traffic and advertising, and general administrative expense. No attempt has been made to estimate and include the cost of pickup and delivery service in these figures.

Total operating costs were estimated by applying these formulas to a series of hypothetic airplane designs. The design studies were complete and were carried out in considerable detail. For example, the design of the engine nacelle was based on the installation drawings of (for the most part) existing engines, and drag factors were computed on this basis. Similarly, landing gear problems were worked out in detail. If the wheel did not go into the wing, the configuration was modified to provide a proper compartment for the wheel. All models are low-wing land monoplanes equipped with a large entrance door. No special loading devices, such as nose or tail doors, have been provided for, but it is believed that if such loading devices were used, all models would be affected similarly and the validity of the comparison would not be impaired. All airplanes provide adequate volume for the maximum usable payload.

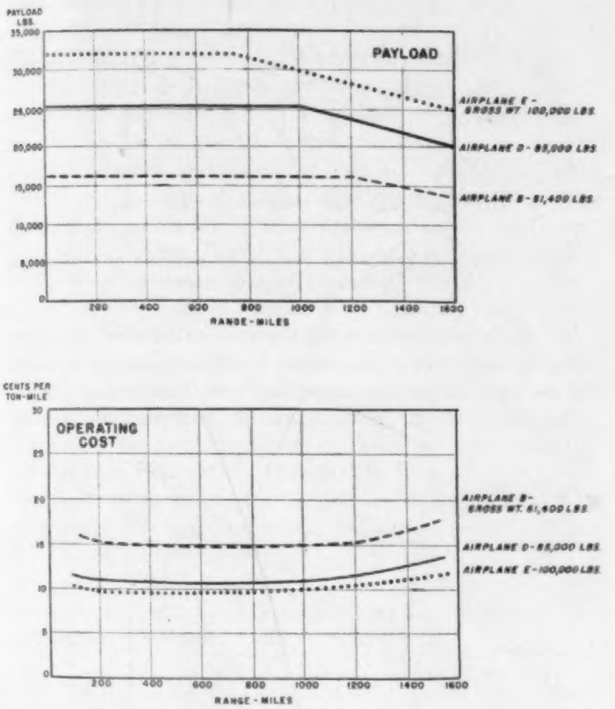
To investigate the economic effect of size, number of engines, range, and speed, various groups of designs were selected, each group providing variation in one of these characteristics. Payloads and operating costs for each series of airplanes were then computed and the results shown in the accompanying charts. A total of 10 airplane designs was investigated, the designs for some airplanes being used on more than one chart. For reference, specifications, payloads, and operating costs for all 10 airplanes have been listed in Table 1. Each airplane has been designated by a letter, and in this paper reference is made to each airplane by the letter designation.

■ The Effect of Size

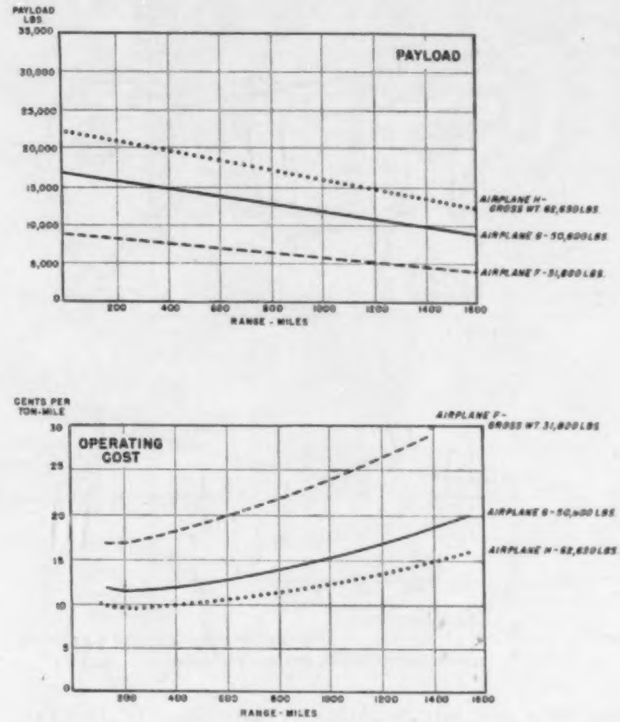
Fig. 1 shows curves for airplanes *B*, *D*, and *E*, and illustrates the effect of airplane size on ton-mile costs in the four-engine series. Airplane *B*, the smallest, has a total horsepower of 4400 and a take-off gross weight of 61,400 lb, while airplane *E*, the largest, has 8000 hp and a take-off gross weight of 100,000 lb. All the airplanes have been designed under the same basic philosophy, resulting in airplanes about halfway between "race horses" and "work horses." Consequently, the variation in ton-mile cost demonstrated in Fig. 1 results almost entirely from the difference in airplane size. It can be noted that, for a variation

Table I—Specifications, Payloads, and Operating Costs
Passenger-Cargo Airplanes

Designation	A	B	C	D	E	F	G	H	I	J
Number of Engines	4	4	4	4	4	2	2	2	2	2
Maximum Normal Power – all engines, bhp	4400	4400	4400	6000	8000	2400	4000	4800	2200	2200
Cruising Speed (50% power), mph	189	185	179	204	219	197	212	212	199	176
Block-to-Block Speed (estimated) (250-mile range), mph	172	160	155	174	184	174	184	182	177	156
Cockpit Crew	3	3	3	3	3	2	2	2	2	2
Take-Off Gross Weight (sea level), lb	53,200	61,400	64,300	85,000	100,000	31,800	50,600	62,650	26,500	33,000
Landing Gross Weight, lb	46,200	53,900	61,600	77,100	92,600	31,800	50,600	62,650	26,500	33,000
Empty Weight, lb	32,450	36,800	38,600	51,000	59,600	22,280	33,170	39,790	17,200	20,858
Wing Span	98 ft	105 ft 8 in.	114 ft 4 in.	126 ft 6 in.	138 ft 6 in.	98 ft	114 ft 4 in.	120 ft	90 ft	120 ft
Wing Area, sq ft	1200	1400	1600	2000	2400	1200	1600	1800	900	1440
Wing Loading, lb per sq ft	44.4	43.8	40.2	42.5	41.6	26.5	31.6	34.8	29.4	22.6
Payload for 1000-Mile Range (50% normal power, no fuel reserve), lb	12,900	16,300	18,350	25,200	30,350	5900	11,600	16,100	5900	8400
Total Operating Expense Per Ton-Mile	$0.1714	$0.1527	$0.1443	$0.1103	$0.1012	$0.2408	$0.1538	$0.1254	$0.2002	$0.1795
Payload for 500-Mile Range (50% normal power, no fuel reserve), lb	12,900	16,300	21,700	25,200	31,950	7400	14,300	19,100	7300	10,000
Total Operating Expense Per Ton-Mile	$0.1714	$0.1527	$0.1221	$0.1103	$0.0961	$0.1920	$0.1269	$0.1057	$0.1690	$0.1508



■ Fig. 1 – Effect of airplane size on payload and operating cost – four-engine airplanes



■ Fig. 2 – Effect of airplane size on payload and operating cost – two-engine airplanes

of 38,600 lb in gross weight, there is (at 500-mile range) a difference of 5.7¢ per ton-mile in operating costs.

From Fig. 2, we see the corresponding data plotted for the two-engine series airplanes, *F, G,* and *H*. It is seen that the same relationship between size and operating cost exists for two-engine as well as four-engine airplanes. 30,850-lb difference in gross weight results in a 8.6¢ difference in ton-mile costs.

These charts illustrate what we believe to be a primary principle in airplane economics: namely, the larger the airplane, the lower the operating cost per ton-mile. This principle is not, of course, of unlimited application. For example, in Figs. 1 and 2, it is apparent that, as size increases, operating cost does not decrease proportionately but moves toward a limiting value. However, it is clear that in the size ranges being considered at present, the largest airplane can operate at a much lower cost per ton-mile than can the smaller airplane.

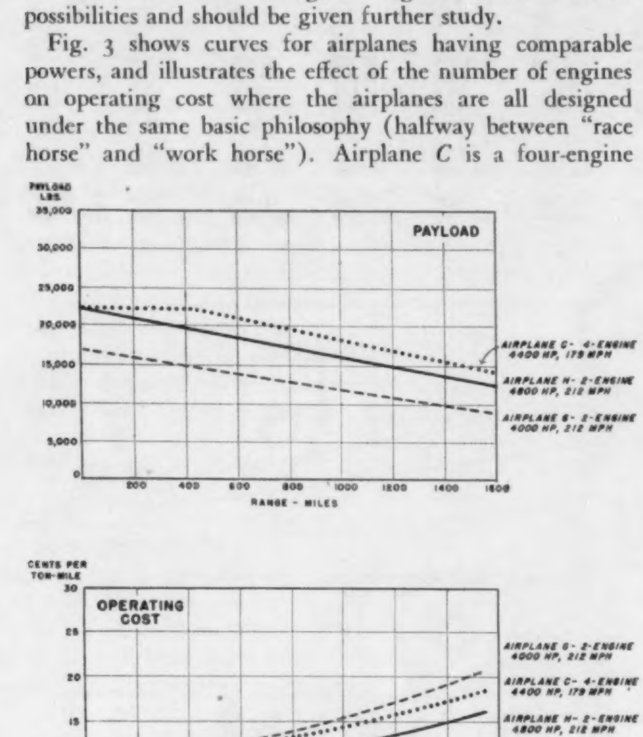
This relationship appears to be due primarily to aerodynamic considerations and, to some extent, to economic considerations. Aerodynamically, the larger airplane is the more efficient since: 1. The diameter of the powerplant decreases with respect to the airplane, as airplane size goes up. 2. The skin-friction drag coefficient decreases as size increases. 3. Protuberances into the air stream, such as door handles, become relatively less important. 4. The ratio of useful load to gross weight increases in the larger airplanes, at least throughout the size range under consideration.

■ Number of Engines

The number of engines designed into each unit has, of course, a direct effect upon operating cost, and this effect was studied by comparing two- and four-engine airplanes having about the same total horsepower. Three-engine airplanes have not been investigated. There are a number of difficulties involved in the designing of the three-engine

airplane. For example, the reduced efficiency of the nose propeller and the increased fuselage drag resulting from the installation of a nose engine are problems which must be met by the designers. In the present study, a number of three-engine configurations were analyzed but none was found which warranted further investigation. However, it is believed that three-engine design offers considerable possibilities and should be given further study.

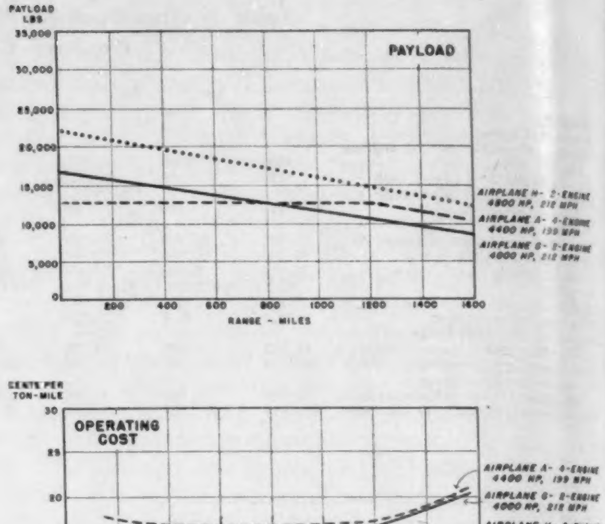
Fig. 3 shows curves for airplanes having comparable powers, and illustrates the effect of the number of engines on operating cost where the airplanes are all designed under the same basic philosophy (halfway between "race horse" and "work horse"). Airplane C is a four-engine



■ Fig. 3 – Effect of number of engines on payload and operating cost – comparable design philosophy

airplane (4400 hp), airplanes G (4000 hp) and H (4800 hp) are twin-engine airplanes. Fig. 3 as well illustrates that, for ranges of 400 miles and over, the cost curve for airplane C, the four-engine airplane, falls about where we would expect a two-engine airplane having the same power to fall, that is, midway between airplanes G and H. The increased payload of the four-engine model is just about sufficient to offset its increased operating cost. However, it should be noted that the cruising speed of airplane C is only 179 mph, while for the twin-engine airplanes it is 212 mph. This loss of speed is primarily a result of the increased engine drag of the four-engine design.

If a comparison between two- and four-engine airplanes is made on the basis of comparable cruising speed, then the results appear as illustrated in Fig. 4. In this figure, airplane A, the four-engine model, has a cruising speed of 199 mph as compared with the speed of 212 mph for the two-engine planes. On this basis, it will be noted that both two-engine airplanes achieve lower operating costs than the four-engine airplanes, over all ranges. And this is true even though airplane G is smaller than airplane A. If their sizes were directly comparable, the margin between G and A would be even greater.

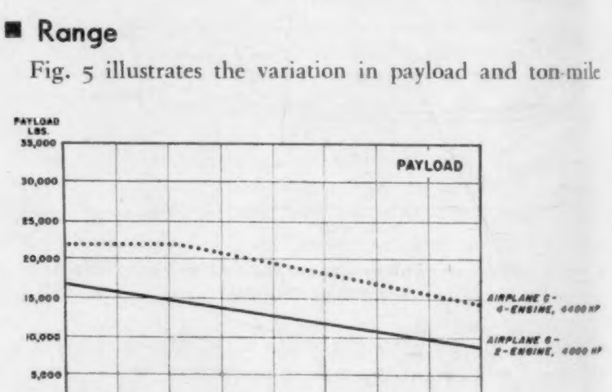


■ Fig. 4 – Effect of number of engines on payload and operating cost – comparable cruising speeds

These charts point toward the conclusion that, for minimum operating costs per ton-mile, where speed is a factor, the twin-engine design cargo airplane is the most efficient where the size of the airplane permits the use of two engines.

■ Range

Fig. 5 illustrates the variation in payload and ton-mile



■ Fig. 5 – Effect of range on payload and operating cost

cost for different operating ranges. A two-engine airplane and a four-engine airplane of the same design philosophy are shown, to illustrate the different range characteristics of each of these models. It will be noted that there is a straight-line relationship between payload and range in the case of airplane *G*, the two-engine model, while with airplane *C*, the four-engine model, the payload curve is flattened at the top. This flattening results from the fact that the allowable take-off weight of the four-engine airplane exceeds its maximum allowable landing weight as specified by the Civil Air Regulations requirements. As a result, the take-off weight must be limited, so that the landing weight is not exceeded for short ranges. The curve for the twin-engine design is not correspondingly flattened, since the requirements for rate-of-climb with one engine inoperative, rather than landing gross weight, are the limiting factor in the design of the two-engine airplane.

The effect on operating costs of these variations in payload can be noted in the operating cost chart of Fig. 5. The cost curves for both airplanes show an upturn at the longer ranges owing to the decreasing payload. However, the curve for airplane *C*, the four-engine design, shows a flatter line along its low portion than does the two-engine airplane. Its low point also falls at a longer range. These curve characteristics result from the flattened payload curve and lead to the conclusion that the four-engine airplane is relatively better able to compete with the twin-engine as a cargo carrier at the longer ranges. From the data at hand, it is not possible to determine what these ranges are for all classes of airplanes, but for the airplanes plotted, in the 4000-4400 hp range, it appears that the twin-engine airplane has its best advantage under 250 miles, whereas the four-engine airplane's position is relatively improved at ranges over that figure. It should be noted also that the lower point in the curve for the four-engine airplanes falls at approximately 450 miles, while the low point in the curve for the two-engine design falls at 300 miles.

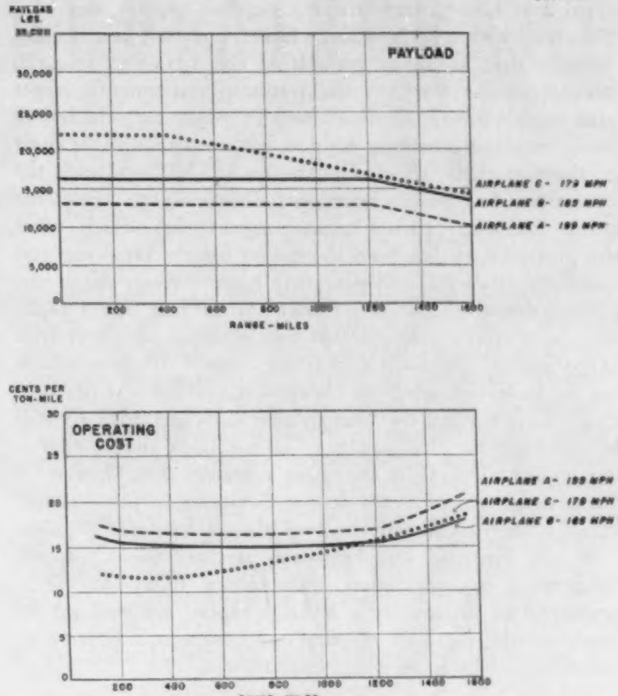
The operating cost curves for both airplanes show an upturn at short ranges, illustrating the increased effect of maneuvering and taxiing time on short trips.

■ Speed

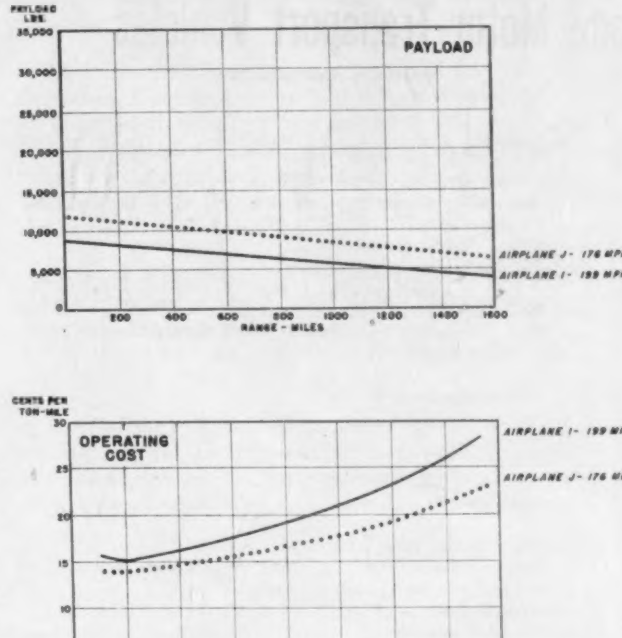
If speed is to be a factor in the development of cargo aircraft within these weights, Figs. 6 and 7 demonstrate the effect on ton-mile costs of variation in cruising speed. Fig. 6 shows a series of four-engine airplanes all designed around an engine installation of four 1100 hp engines. Wing loadings and gross weights of the individual airplanes were varied, resulting in variation in cruising speeds from 179 to 199 mph. From the operating cost chart it can be noted that, at ranges of 500 miles, the 20-mph difference in speed resulted in a difference in operating cost of almost 5¢ a ton-mile. And, it is interesting to note that at ranges of over 1200 miles, airplane *B*, of medium cruising speed, has the lowest operating cost.

In the two-engine series plotted in Fig. 7, there is also a difference in cruising speed of approximately 20 mph. In this series, however, the variation in operating cost is considerably smaller, varying, at 500 miles, from 17¢ per ton-mile for the faster airplane to 15¢ for the slower.

Therefore, it may be stated that in a series of four-engine cargo airplanes, similar to those under discussion here, speed must be paid for on each ton at the rate of about ¼¢ per mph. In two-engine airplanes similar to those studied, speed is worth about 1/10¢ per mph. Whether or



■ Fig. 6 – Effect of speed on payload and operating cost – four-engine airplanes



■ Fig. 7 – Effect of speed on payload and operating cost – two-engine airplanes

not this price is justified is not within the scope of this paper but is a matter which will have to be determined by the prospective users of these airplanes.

In conclusion, first, it appears that our domestic markets in the immediate post-war era, from an economic standpoint, seem to fall in the following order by virtue of

tariff and speed competition: One, rail express and first-class mail within certain range limits; and two, l.c.l. freight. Second, that as an outgrowth of the advanced research because of the war and the resultant refinements, larger and more efficient airplanes will be produced which will bring airplane operating costs to within a reasonable limit of these markets; third, that the trend will be toward the operation of large combination passenger-cargo planes for trunk line and primary connecting carriers which, after the groundwork has been developed for the servicing and handling of cargo, will give way to exclusively cargo airplanes; that the secondary feeder and off-line feeder plane will be a convertible efficient combination passenger-and-cargo plane; and fourth, that the largest airplane which can be built will offer the cheapest operating cost per ton-mile. This necessarily assumes that sufficient payload will be available for this airplane. It has been shown that a two-engine airplane is the most economical to operate if the payload requirements do not dictate an airplane larger than can be handled by the two biggest available engines.

As the airplane was originally devised as a peaceful medium of transportation, although it today has been developed as principally a lethal weapon, we will see its return to the use of its original conception, and peace and commerce will ride its wings tomorrow.

Lubricants for Ordnance Combat and Motor-Transport Vehicles

continued from page 349

3. Gravity, API at 60 F ....
4. Sediment and water ...... Maximum ...... % by Volume
5. Viscosity at 100 F, SU (Sec) ....
6. Carbon Residue, % by weight ....
7. Ash, % by weight ....
8. Cetane Number ....
9. Sulfur, % by weight ....
10. Distillation, F:
    IBP ..; 10% ..; 50% ..; 90% ..; EP .. Recovery; .. %

B. Lubricating Oil Characteristics
1. This lubricant is designated by our Formula Number .....
2. Viscosity grade SAE .... V.I. ....
3. Viscosity, SU (Sec) at: 100 F ....; 130 F ....; 210 F ....
4. Carbon Residue, % ....
5. Sulfur, % ....
6. Phosphorus, % ....
7. Chlorine, % ....
8. Ash, % ....
9. Other identification tests ....

C. Engine Test Identification
1. Test was run at .... under their oil code No. ......
2. Using Applicant's Lubricant Formula No. ....
   Grade SAE ....

D. Evaluation of Test Results
1. For low carbonaceous deposits
   a. Amount and nature of carbon deposit on liner above ring travel ....
   b. Amount and nature of deposits around piston crown ....
   c. Piston-crown scuffing (nature and quantity) ....
   d. Amount and nature of deposits behind rings ....
   e. Nature of deposits on sides of rings ....
   f. Nature of deposits on sides of grooves ....
   g. Nature of deposits on ring lands ....
   h. Skirt condition ....
   i. Underside of piston ....
2. For high film strength
   a. Liner condition (scratched or not scratched) ....
   b. Number of compression rings scratched ....
   c. Piston-ring sharpness ....
3. For resistance to ring sticking
   a. Number of tight rings ....
   b. Number of stuck rings ....
4. For insurance to open oil channels
   a. Deposits in oil ring slots ....
5. For cylinder wear
   a. Wear on cylinder diameter 1¼ in. down from top
      (1) Transverse ....
      (2) Longitudinal ....

E. General Engine Test Operating Conditions

	High	Low	Average
1. Brake horsepower	....	....	....
2. Oil to bearings, temperature, F	....	....	....
3. Water outlet temperature, F	....	....	....
4. Blowby of gases in crankcase, cu ft per hr	....	....	....

5. Lubricating oil consumption for 6-126 hr period .. hr per gal
   126-246 hr period .. hr per gal
   246-366 hr period .. hr per gal
   366-486 hr period .. hr per gal
6. Average lubricating oil consumption
7. Remarks ....

F. Photographic Evidence
1. Show by photograph the nature of deposits found on test pistons in accordance with instructions set forth in the test procedure.

G. Engine Parts, Evidence
1. The pistons and liner from this test were shipped to Armour Research Foundation .... for further examination. (date)
   This questionnaire is a part of the Final Test Report to which it is attached.
   Date .... Signed ....
   Company ....

Cold-Starting Tests on Diesel Engines

continued from page 361

in cranking speed and ambient temperature and decrease in altitude.

2. The starting performance of undoped fuels is predicted by the delay cetane number. With increased cetane number, greater ease of starting is obtained. However, with doped fuels, the delay cetane number may or may not predict the starting performance.

3. As indicated by tests on one make of engine, laboratory results may be used to predict service starting performance.

4. Various substances, including chlorine, hydrogen sulfide, amyl nitrate, ethyl disulfide, and chloropicrin, are effective in aiding starting when added to the intake air. This finding suggests the use of an auxiliary applicator, especially designed for the purpose, for aiding in starting under severe weather conditions. However, the effect of such dopes on engine maintenance should be determined.

PROBLEMS in WOOD AIRCRAFT

by IVAR C. PETERSON*

OUR present effort to turn out wood aircraft in mass-production quantities and at the same time to take full advantage of recent developments in glues, fabrication technique, and design has brought us face to face with many new problems as well as some skeletons of the past. Although it is not possible to give the answer to any of these problems with either the degree of thoroughness or accuracy that is desired, a partial answer at the present time may be of more value than the complete answer at a later date.

For purposes of discussion and, it is hoped, for convenience in use, the material which follows has been grouped essentially under the three main headings of fabrication, static testing, and detail design. The various topics which are covered by no means comprise all of the "problems in wood aircraft," and it may be that a number of these items already have been explored more thoroughly by certain aircraft manufacturers or research organizations than is indicated by the information contained in this paper.

Before entering into a discussion of the various problems confronting wood aircraft manufacturers, it might be well to point out a few of the strength peculiarities of wood that are the basis for some of our troubles.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 8, 1943.]

* At time of presentation, aeronautical engineer, Civil Aeronautics Administration; now, secretary, Airplane Technical Committee, Technical Department, Aeronautical Chamber of Commerce of America, Inc.

■ Peculiarities in Wood Strength Properties

It is often not understood just why wood strength tables fail to give values for tensile strength, although the tensile

PRESENT-DAY efforts to produce wood aircraft in large quantities have uncovered many new problems, for wood has certain peculiarities that must be taken into consideration by the engineer, if he is to design structures that make full use of the benefits to be derived from wood.

Attempts to take advantage of the high tensile strength of wood will lead to failures in shear, because loads theoretically in tension practically always have shear components that are great enough to overcome the low shear strength of wood. Moisture content also has a great effect on the strength of wood; and the moisture equilibrium of a piece of wood will vary with the relative humidity and temperature to which it is exposed.

Mr. Peterson discusses some of the problems confronting the wood aircraft manufacturer under three headings: fabrication, static testing, and detail design.

The fabrication of wood structures revolves around the production of strong glue joints. Sufficient glue spread on surfaces that have been carefully smoothed, and a correct balance between temperature, pressure, and glue consistency at the time pressure is applied are the basic requirements.

The problem of "reducing" static-test data for wood structures is very difficult and cannot be associated with the more-or-less standard correction procedures that have been established for metal structures. Certain factors that affect the strength of wood tend to affect one another, and corrections to static-test data for complete wood structures are unnecessary, provided some control is maintained over the specific gravity of the material in the test article.

Lack of knowledge of plywood strength and elastic properties, and poor detail design practices are responsible for the somewhat poor reputation that wood aircraft structures have acquired recently.

Both of these items will be overcome as a background of design information and experience is obtained similar to that already available for metal structures.

★ ★ ★

THE AUTHOR: IVAR C. PETERSON, who several months ago resigned from the Washington office of the Civil Aeronautics Administration to accept a position in the Technical Department of the Aeronautical Chamber of Commerce of America, Inc., as secretary of the Airplane Technical Committee, was formerly aeronautical consultant and assisted in preparation of the "ANC Handbook on the Design of Wood Aircraft Structures." Following his graduation from the University of Washington, Seattle, in 1938 with a B. S. degree in aeronautical engineering, Mr. Peterson worked in the detail-design group at North American Aviation, Inc., and for two years was employed in the Structures Laboratory at the National Advisory Committee for Aeronautics at Langley Field, Va.

strength of wood parallel to the grain is very high, being about 18,000 psi for straight-grained spruce. Thus, the ratio of tensile strength to specific gravity for spruce is approximately 45,000 as compared to 23,600 for 18-8 stainless steel in the full-hard condition.

The reasons why this high tensile strength cannot be utilized in structures are:

1. The low shear strength of wood.
2. The highly critical effect of stress concentrations and slope of grain on tensile strength.

It is practically impossible to obtain pure tension stresses in a member, or to devise fittings that will permit the tensile strength of the full cross-section to be developed and, as a result, shear failures result rather than tension failures.

Since the permissible slope of grain in aircraft lumber is 1 in 15, the allowable tensile strength must be set accordingly. For spruce, a grain slope of 1 in 15 will reduce the straight-grained tensile strength to about 9400 psi. A similar reduction was also found to be true for other species and, because these tensile strengths (for a 1 in 15 grain slope) checked the modulus of rupture in static bending quite closely, the latter property has been conveniently used to represent the allowable tensile strengths parallel to the grain. In contrast to the high tensile strength of Sitka spruce parallel to the grain, the specific strength in tension perpendicular to the grain is only 600 psi, and in shear parallel to the grain, 2500 psi, as compared to 23,000 and 16,000 psi, respectively, for 18-8 full-hard stainless steel. (These values are approximate.)

The strength properties of wood are very much affected by changes in moisture content and, unfortunately, the moisture equilibrium of a piece of wood will vary with the conditions of relative humidity and temperature to which it is exposed. Near Seattle, Wash., and other humid regions, the wood moisture content averages approximately 15%, while in certain regions of the Southwest, a moisture content of 5% is not uncommon. The differences in strength of Sitka spruce for two moisture contents, 15% and 7%, are shown in Table 1.

Table 1 – Comparison of Sitka Spruce Strength Properties at 7% and 15% Moisture Content

Strength Property	Average Strength at 7% Moisture Content, psi	Average Strength at 15% Moisture Content, psi	Increase in Strength from 15% Moisture Content to 7% Moisture Content, %
Static Bending Proportional Limit	8950	6200	45
Static Bending Modulus of Rupture	12800	9400	36
Maximum Compression Perpendicular to Grain	1180	840	40
Maximum Compression Parallel to grain	7600	5000	51
Shear Strength Parallel to Grain	920	750	23

From the standpoint of efficiency and avoidance of shrinking and swelling difficulties, it would be desirable if wood aircraft could be designed for operation in a particular geographical location. For example, a spruce wing spar of 1 x 6-in. cross-section (designed on the basis of 15% moisture content) could be reduced to slightly less than a ¾ x 6-in. cross-section on the basis of a 7% moisture content, with a resulting weight saving of 28%.

Gluing and Fabrication Problems

Gluing Recommendations – Most glue companies supply general instructions on gluing temperatures, pressures, spread, and assembly periods, which the aircraft manufacturers are advised to follow in order to obtain the best results from a particular glue. To supplement these glue manufacturers' instructions, a number of recommendations are given in Table 2.

Final Assembly of Molded Shells to Framework – One of the problems associated with the final gluing of molded plywood shells to bulkheads, ribs, or other framework, is that of getting a true fit between the bulkhead contours and the inner surface of the shell. Since these molded shells usually have double curvature, they are quite rigid, and a misalignment of the mating surfaces will result in a poor glue bond regardless of the amount of pressure applied. When this shell type of construction is used, the necessity for prefitting and maintaining close tolerances must be impressed upon the shop personnel.

Assembly-Period Requirements – With the above-mentioned type of construction, manufacturers are having difficulty in meeting the "assembly time" requirements with cold-setting glues. For instance, it is very desirable to mold plywood shells in large units, but when these shells are to be glued to a structural framework it becomes a problem to spread the glue, fit the shell into place, and provide the necessary clamping pressures all within a period not to exceed 15 or 20 min. Possible solutions to this difficulty are:

1. A well-planned assembly routine with plenty of coordinated help.
2. A reduction in the size of the molded shell units.
3. Use of a low-temperature phenolic-resin glue which permits considerably longer assembly periods than do the cold-setting glues.

Duration-of-Pressure Requirements – The pressing or setting time necessary for casein or cold-press synthetic-resin glues range from a minimum of 3 hr to 6 hr or more under normal room temperatures. This means that jigs are tied up to such an extent that from 50 to 75% more tooling is necessary to reach the production that would be possible if pressing periods were not required. What is desired, of course, is some means of accelerating the setting so that a higher proportion of the final glue strength is developed before the pressures are released and the assembly removed from the jig.

A short pressure period is favored by a concentrated glue; a thin spread; warm, dry, thick layers of wood; and a warm room. The most effective way to hasten setting is to increase the temperature.

High-frequency heating has received considerable publicity, as this would permit even the use of hot-press resin glues, but the original cost of the equipment and certain material shortage put this method out of reach for the present. (It might be well to mention at this time that some of the recent wild claims for tremendous strength increases in wood structures through the use of high-frequency heating are entirely without basis. High-frequency heating contributes nothing to the basic strength of wood.)

Hot-rooms have been used to advantage by some manufacturers in reducing pressing periods, but there are several

Table 2 – Tips on Gluing Practices

1. Desirable wood moisture content at time of gluing:
   (a) For cold-press gluing.
   Thickness under $\frac{1}{8}$ in. – 5 to 8%.
   Thickness over $\frac{1}{8}$ in. – 8 to 12%.
   (b) For hot-press resin gluing (water or water and alcohol suspensions).
   Thickness under $\frac{1}{16}$ in. – 5 to 8%.
   Thickness over $\frac{1}{16}$ in. – 8 to 12%.
   (c) For hot-press resin gluing (dry film, or alcohol suspension).
   All thicknesses veneer – 8 to 10%.
2. Normally recommended gluing pressures should be increased when:
   (a) Laminations are thick and surfaces are warped.
   (b) Woods are hard and dense and a more viscous glue is used.
3. Automatically maintained fluid pressure, dead weights or calibrated spring pressure are preferable to ordinary clamp pressures or nail pressures that have no automatic "follow up" in maintaining the original required pressure. Wood yields under pressure, the glue line becomes thinner, and under heat the wood will lose moisture and shrink.
4. Clamps or other pressure devices should be closely adjusted for a short time after the initial application of pressure in order to insure that the proper amount is maintained.
5. When relatively thin pieces are being glued, blocks and cauls should be used to afford an even distribution of pressure. Such blocking is desirable in all cases if local crushing of the wood is likely to occur.
6. Nail gluing should not be used with constructions thicker than about $\frac{1}{8}$ – $\frac{3}{16}$ in. Cement coated, Nos. 18 or 20 gage nails are recommended, and the spacing should be such that each square inch of surface contains at least one nail.
7. Spreads much higher or much lower than those normally recommended are undesirable, but an excess is safer than an insufficient amount.
8. Normally recommended glue spreads should be increased when:
   (a) Rough or porous surfaces are to be glued.
   (b) Unusually dry wood is to be glued.
   (c) Assembly periods are long.
   (d) Pressures are obtained by nail gluing.
9. Under adverse conditions of gluing, such as when the glues become very thick before pressing, double spreading is more reliable than single spreading.
10. When a low density wood is glued to one of high density, a glue mixture somewhat thicker than normal and gluing pressures approaching the maximum the lighter wood will stand should be used.
11. When gluing dense hard woods with casein glue, and when high pressures are to be used, a more viscous or thicker glue mixture should be used so that too much of the glue will not be squeezed out, thus starving the joint.
12. With cold-press glues very short assembly periods (less than 3 min) should be avoided unless the pressure is reduced somewhat.
13. Normally recommended assembly time should be decreased when:
    (a) A thicker or more viscous glue is used.
    (b) Moisture content of wood is low.
    (c) Temperature of either wood or workroom is high.
14. In no case should the working room or wood temperature ever go below the minimum temperature recommended by the glue manufacturer when cold setting urea-resin glues are being used.
15. With normal wood, smooth, even surfaces produced on planers and jointers are best for gluing. Roughening the surfaces by sanding, or wire brushing, is recommended only for preparing "glazed" surfaces, areas showing bleed-through, areas affected by squeeze-out around glue joints, and surfaces not readily wetted by water due to waxes, oils, or other foreign material. (Glazed surfaces can often be brought to near normal condition by wetting with water and bulk piling then reconditioning before gluing.)
16. Sawed surfaces made with a dull saw will give a poor glue bond as a result of damaged wood fibers. Although the gluing of sawed surfaces is not recommended, fairly good results can be obtained if a clean cut is made and all loose dust particles are carefully removed before gluing.
17 When gluing oily or resinous species such as pitch pine, red and white oak, hard maple, black cherry, and sweetgum, best results can be obtained if the wood surfaces are given a light machine cut just prior to the gluing operation.
18. Plywood and laminated wood floats and flying boat hulls which may be submerged in sea water for long periods of time should be fabricated with phenolic resin glues. Salt water causes deterioration of urea-resin glues and either salt water or fresh water causes deterioration of casein glues.
19. Wood which has undergone acidic fire-proofing treatment may be glued with phenolic-resin glue but information is not available as to the reliability in this case of casein or urea-resin glues.
20. Casein glue is recommended for making field repairs when gluing conditions cannot be controlled, since fairly good results can be obtained under a wide range of wood moisture contents (3 to 16%) and gluing temperatures (as low as 40 F).
21. "Sizing" of scarf joints or other end-grain joints is recommended, particularly for these denser hardwoods.
22. When sizing with casein or phenolic-resin glues, the "size coat" should be allowed to dry thoroughly before the final glue operation. For urea-resin glues, some glue manufacturers recommended that the "size coat" be allowed to dry only partially before the final glue operation.
23. When applying the necessary pressures to scarf joints there is a strong tendency for end slippage to occur. If satisfactory scarf joints are expected it is essential that every precaution be taken to prevent such slippage.
24. Unmixed glues, particularly the resins, should be stored in a cool place.

factors which must be taken into consideration. Additional floor space is required if the assembly operation is not made in the hot-room itself, and if metal jigs are used and close tolerances are involved, the increase in temperature will result in undesirable dimensional changes. Humidity control as well as temperature control is necessary to maintain the wood or plywood moisture content equilibrium. Although it is normally assumed that a temperature increase from 70 F to 90 F will cut the total pressing period in half, this will be true only for relatively light constructions. When heavy members or thick plywood sheets are involved, additional time must be allowed for the additional heat actually to penetrate to the glue line.

Some experimenting has been done with the use of infra-red lamps for reducing setting periods. The results to date, however, are not particularly encouraging. One disadvantage results from the lack of humidity control of the air immediately surrounding the wood surface; consequently excessive drying out takes place. Although high surface temperatures can be obtained by the use of infra-red lamps, this heat has a unique characteristic in that it is not readily transferred through the wood to the glue line, due possibly to the low capacity of the source. In other words, a large differential will exist between the temperature at the glue line and that at the surface. For this reason, manufacturers should be very cautious about reducing their pressing periods on the basis of surface temperatures.

Increased use is being made of various types of electric heating strips and, from the standpoint of facility in application, these heating strips appear to be the most promising method of reducing the pressing periods for cold-setting glues. Because of the low cost involved, it is possible to develop various shapes and sizes of heating strips or pads to meet each particular application, and with voltage control the temperature can be maintained at a fairly constant value. It is quite simple to obtain very high temperatures with these electric heating strips and, to cut pressing periods to a bare minimum, some manufacturers are inclined to go to temperatures well over 200 F. It should be remembered that cold-press glues were developed to set at temperatures of about 75 F to 85 F. Although accurate information is not yet available, indications are that high setting temperatures with cold-press glues may result in joints that will retain their strength for only a short time.

Creep in Thermoplastic Glues – A number of thermoplastic resin glues have been developed which appear to have all of the desirable characteristics of good bag-molding or final-assembly glues, but manufacturers should not use these glues in highly stressed parts of the structure. Thermoplastic glues are subject to "creep" at elevated temperatures (approximately 160 F). Since wood surfaces coated with a dark camouflage paint may reach temperatures well over 200 F in desert regions, separation of the glue joints is likely to occur.

Gluing of High-Density Materials – Glue manufacturers in general will not recommend or guarantee their glues for use in gluing dense materials, such as compreg or plastic laminates, and it becomes necessary for aircraft manufacturers to rely upon rather meager data resulting

from their own experience with certain types of glue. A few sources of information, however, indicate that reasonably good results can be obtained in gluing compreg to compreg or plastic laminate to plastic laminate if a low-temperature phenolic-resin glue is used and the surfaces are roughened slightly.

When compreg is glued to normal wood, it is recommended that a layer of impreg be inserted between the two materials. The primary objective is to reduce as much as possible the stress concentration in any single glue line due to pronounced differences in moduli of elasticity.

Nailing Strips Versus Permanent Nailing – The primary function of nails in nail gluing is to bring the joining surfaces into intimate contact with one another while the glue sets. Because of the more even distribution of pressure possible for an equal number of nails, nailing strips are much preferred to permanent nailing. Further disadvantages associated with the use of permanent nailing are that of the nails working themselves out and spoiling the finish and the considerable increase in structural weight due to the weight of the nails themselves. Needless to say, these nails add nothing to the structural strength.

■ Static Testing of Structural Elements

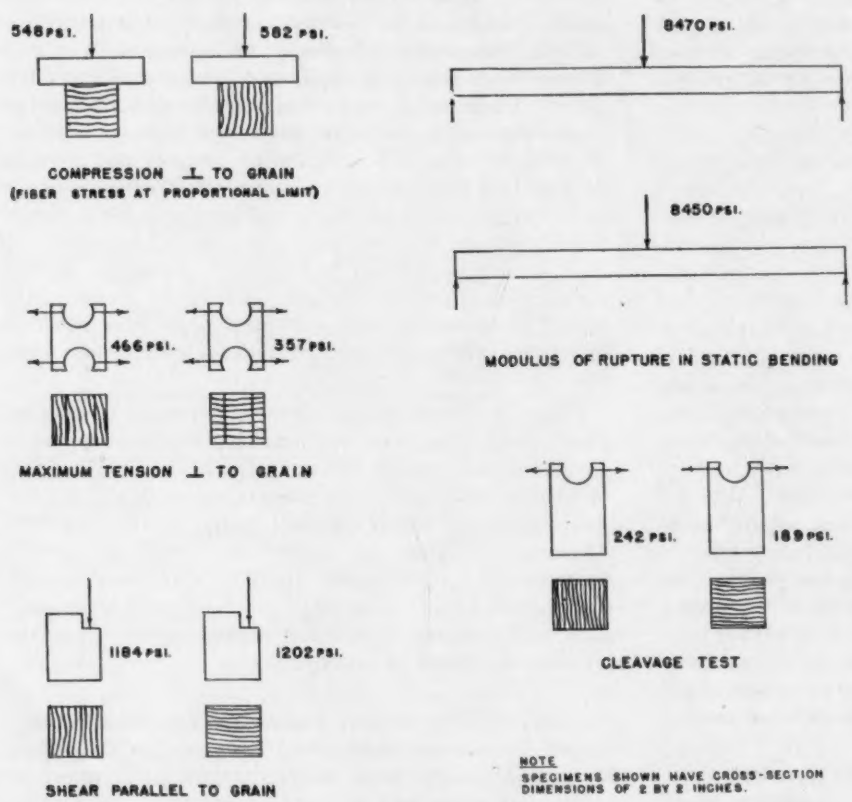
Position of Growth-Rings in Test Specimens – In the sawing of lumber the position of the growth rings may be made to assume different directions with respect to the surface of the piece.

In making test specimens, and also in analyzing certain static-test data, it is often wondered just what practical significance can be assigned to the position of the growth-rings relative to the direction of loading. The sketches in Fig. 1 serve to illustrate the quantitative effect that growth-ring positions has on the strength properties of Sitka spruce. The results are for specimens of 2 x 2-in. cross-section. Significant differences with ring placement may become evident in properties not appreciably affected in the 2 x 2-in. pieces when specimens of smaller size are tested.

Plywood Test Specimens – The test procedures and specimen sizes have been more or less standardized for most common structural materials, but the use of these "standard" specimens has not been satisfactory for tests on plywood elements, in many cases. The main difficulty encountered in tests on plywood elements is that the type of failure which occurs is often something other than that desired. For this reason, it is necessary to develop new test procedures and specimens that will provide test data from which suitable design allowables can be derived.

Plywood tension specimens and flexural fatigue specimens having the face-grain direction parallel to the applied load should be notched or tapered very gradually. If the tension stresses are transferred too rapidly because of a sharp taper, shear failures will always occur at low loads.

Plywood tension specimens having the face-grain direction at angles other than 0 deg (parallel) to the applied load should neither be notched nor tapered in the shank length. The specimens should also be made wide enough to ensure that shear failures along the grain will not occur. For 45-deg plywood tension tests it has been found, in some cases, that a specimen of 6-in. width and 6-in. shank length is necessary for satisfactory results.



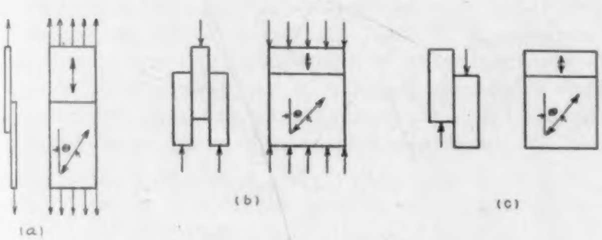
■ Fig. 1 – Average results of tests showing the influence of growth-ring position on spruce strength properties

Plywood flexural (beam) specimens having the face-grain direction other than parallel or perpendicular to the applied stress should be sufficiently wide to cause the deflection to be constant across the beam at any point in the span length. If the bending modulus of elasticity is being determined experimentally, a uniformly distributed chordwise load should be applied and the specimen restrained so that it will not twist and leave the supports.

Block-compression tests in which the plywood face-grain direction is other than parallel or perpendicular to the applied stress will not give consistent results unless the specimen width is greater than its height. For 45-deg plywood block-compression tests, it is suggested that the ratio of specimen width to height be made approximately equal to two.

Establishing Glue-Joint Shear Allowables by Static Tests - The question of allowable stresses for glued joints is quite intimately related to the shear strength of wood and, since this happens to be one of the properties in which wood is relatively weak, the sizes of various members in aircraft structures are often dictated by glue-area requirements. The extreme difficulty encountered in trying to set up standard test methods to obtain satisfactory glue-joint shear allowables is the reason why more comprehensive information on this subject is not already available.

It is safe to say that for each glue shear-test method which can be devised (see Fig. 2) there will be a corresponding number of different test values. This explains why it is difficult to reduce such test data to a basis from which reliable design values can logically be derived. The amount of area in shear, the eccentricity of the applied loads, the species, the type of joint, and the method of testing all have an effect upon the quantitative results obtained.



■ Fig. 2 – Methods for making glue-joint shear tests

In section 2.911 of Supplement No. 1 to the ANC Wood Handbook, glue joints have been placed in three classes:

1. Joints having parallel-grain gluing.
2. Joints having cross-grain gluing.
3. Joints not conforming to either of these two classes.

(It should be noted that any joint involving plywood cannot be regarded as a class-1 joint. Even though the face-grain direction of the plywood is parallel to the grain of the adjacent piece, it is obvious that the glue joint between the first and second plies is cross-grain, therefore the critical joint.)

The allowable stress for joints which have parallel-grain gluing may be taken equal to the ultimate shear strength parallel to the grain for the weaker species. Static tests frequently give glue shear strengths considerably higher than the shear strength of the wood itself, but even if these strengths were valid it would not permit a reduction in the required joint area. In other words, shifting the critical section from the glue line to the adjacent wood fibers will not permit any reduction in the area required to carry the shear stresses.

The allowable stress for joints which have cross-grain gluing may be taken equal to one-third the ultimate shear strength parallel to the grain for the weaker species. Fifty per cent of this reduction has been found necessary to allow for the "rolling shear" type of failure of the wood fibers, and the other 50% to allow for stress concentrations in most types of joints. It is possible to say, therefore, that cross-grain glue joints with small eccentricities and a relatively uniform shear distribution could be allowed two-thirds rather than one-third of the ultimate shear strength of the wood.

There is no standard glue-joint shear test by which the allowable value of "one-third the ultimate" for cross-grain gluing can be verified, although the plywood glue-shear tests required by the plywood specification (AN-NN-P-511) yield results which are quite comparable to this value. It has also been found from experience with tests on full-scale structures that glue-joint allowables established in accordance with the above information are satisfactory.

It is reasonable to expect a general upward trend in the strength of glued joints when the relative grain direction of the adjoining pieces is varied from cross-grain to parallel-grain. An allowable shear curve is given in Fig. 3 for use in the design of glue joints which have the grain direction of one piece at various angles to the grain direction of the piece to which it is glued. This allowable or design curve was derived by first running a number of glue-shear tests at various angles to establish a quantitative trend, and then setting the maximum value (at $\theta = 0$ deg) equal to 100% of the ultimate wood shear strength for the species tested. In this manner the design glue-shear curve was made applicable for all species or combinations of species.

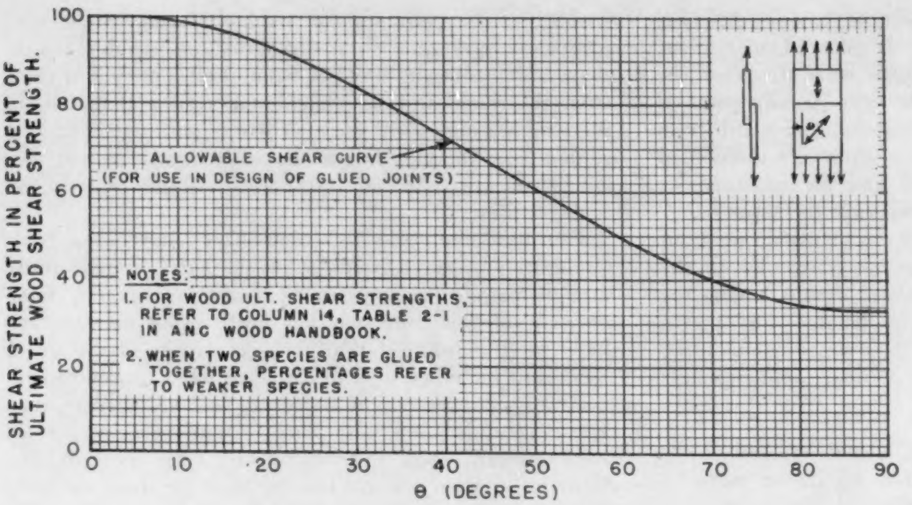
■ Problems in Reduction of Static-Test Data

It is not the purpose of this paper to become involved in a discussion of the various test loads required to substantiate the strength of wood aircraft structures, since these loads are specified by the approving agency.

What is of concern at this time, however, is the increase in static-test loads above the normal design values due to certain correction factors. These factors are *not* "additional factors of safety" but rather "adjustment factors" to take into account the effect of strength variations of the material. The adjusted static-test data are thus supposed to represent the minimum strength that a particular structure would have if it were constructed entirely of material just meeting the minimum required strength properties.

In the case of metal structures, various methods have been established for "reducing" static-test data to minimum guaranteed properties, but the reduction of static-test data for wood and plywood structures is considerably more complicated than for metals. The inherent variability of wood itself makes any method of correction, at best, only an approximation.

Corrections to wood strength and elastic properties can be placed under the headings of (1) specific gravity, (2) moisture content, and (3) duration of load.



■ **Fig. 3 – Variation in glue-joint shear strength with relative grain direction of glued pieces**

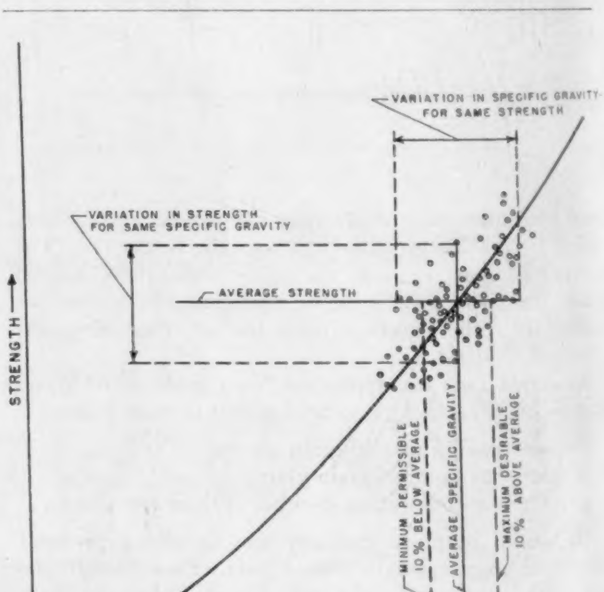
Specific-Gravity Corrections – Unfortunately, the term "minimum guaranteed properties" is not applicable to wood and plywood structures and, as a result, the current attempts at correction of static-test data have led to nothing but confusion. It is true that a "minimum permitted" specific-gravity value is specified for all aircraft lumber, but the design allowables for these species are related to "average" specific-gravity values rather than to this "minimum permitted." (Reference columns 2 and 3, Table 2-1 in ANC Wood Handbook.) As shown in Fig. 4, the magnitude of the average specific gravity has been determined by a consideration of values both above and below this so-called average. It is thus apparent that any strength property for a given species should be associated with a "range" of specific-gravity values rather than an average, and that both the upper and lower limits of this range should be specified in any wood strength table. Since the minimum permitted value was placed 10% below the average, it is reasonable arbitrarily to set a maximum value at 10% above the average. (Thus, for spruce the specific-gravity range should be 0.36 to 0.44.) This upper limit need not be considered as the maximum permitted but rather as the maximum desired in order to avoid excessive structural weight.

Another point that should be noted in Fig. 4 is the considerable variability in wood strength properties for identical values of specific gravity. This variation is typical of all of the wood strength and elastic properties, some to a greater degree than others, but the overall average of this variation is approximately 10%.

In view of the above points, it is believed that static-test data for isolated wood structures, either major assemblies or component parts, should not be corrected for variation in specific gravity. In connection with this policy, however, the aircraft manufacturer should select material that has a specific gravity between ±10% of the "average" value specified in order to obtain results which are indicative of the average strengths.

When a series of correlated tests on component parts is being made, correction of static-test data to the average specific gravity for the species will greatly reduce the overall scatter of test points. These corrections can be made by use of the correction constants and formula given in Table 2-3 of the ANC Wood Handbook. If the specimen is made of both wood and plywood, such as stiffened plywood panels, the above specific-gravity correction need be made only for the solid wood parts. When the effect of a particular variable is being determined from a comparison of tests on several specimens and any degree of accuracy is desired, it is essential that the specimens be matched as closely as possible; that is, constructed from the same plywood sheet or the same piece of wood stock (end to end or tangential).

Since the veneer cutters maintain no control over the weight of veneers which go into making up plywood panels, it is very impractical to attempt any specific-gravity corrections to plywood static-test data. Specimens should be fabricated from plywood that is fairly representative in both *weight and thickness* of the plywood to be used in the actual aircraft structure. In the past, considerable variation was found to exist between the weights and strengths



■ **Fig. 4 – Variation of the strength of wood with its specific gravity**

of plywood purchased from different plywood manufacturers under identical specifications.

Moisture-Content Corrections – The allowable strength and elastic properties of wood and plywood for use in aircraft design have been based on a moisture content of 15%. Because of the effect of change in moisture content upon the strength properties, it is sometimes necessary to correct static-test data for wood members having a moisture content other than 15%. For purposes of discussion, the problem can be considered in two parts: (a) Strength tests of major assemblies; (b) Tests of structural elements.

(a) Strength tests of major assemblies

Test data for major wood and plywood assemblies, such as wing structures and fuselage sections, cannot usually be corrected for changes in moisture content within any reasonable degree of accuracy for the following reasons:

(1) The magnitude of the correction factor is dependent upon the type of stress that causes failure, but this is not so easy to judge. Wood structures often fail so suddenly and so completely that the exact cause of failure is difficult or impossible to distinguish from the results of failure.

(2) A combination of stresses usually exists in the members, but moisture-strength relations have been determined for only the simple cases of pure stress.

(3) The effect of change in moisture content upon the tensile strength of wood has not yet been determined, but rather meager test data indicate that this effect is quite small.

(4) Factors contributing to the failure are entirely hidden; that is, changes in moisture content of the entire structure have a definite effect upon the deflections and the stress at which failure occurs, but this effect cannot be evaluated.

(5) There is usually a wide moisture differential between the solid wood members and the plywood parts of the same airplane. Therefore, correcting the actual test strength of both the wood and plywood parts to the same moisture content of 15% will upset the balance of strength that existed between them at the time of test.

(6) The critical buckling strength of plywood, such as instability failures, is not affected by changes in moisture content. The reason for this is that the change in modulus of elasticity counteracts the change in the plywood thickness. The Forest Products Laboratory's preliminary tests show that the value of Et^3 of the plywood is practically constant over a wide range of moisture content.

Although the strength of a wood structure depends upon its moisture content, the above points indicate that any rational method for correcting static-test data for variation in moisture content would lead to unending difficulties. Moisture corrections to static-test data could best be handled by the use of an arbitrary correction factor. Since the moisture content of most aircraft structures is less than 15%, this factor (multiplying) would be something less than 1.0. However, on large assemblies the test load is usually sustained for a considerable period of time. This duration-of-load effect can be considered to offset the effect of a somewhat low moisture content of the wood. For this reason, it is believed that moisture corrections to static-test data for *complete structures or major assemblies* need not be made under normal conditions of testing.

(b) Tests of structural elements

Moisture-content corrections to static-test data for solid wood parts such as compression specimens, shear specimens, members subjected to bolt-bearing loads, flexural specimens, should be made by means of the constants given in Table 2-2 in the ANC Wood Handbook. This table covers all wood strength properties with the exception of tension, and until additional information becomes available the effect of moisture-content changes on the tensile strength of wood may be neglected.

Moisture-content corrections to static-test data for plywood parts are more difficult to make than for solid wood because of the different directions in which the adjacent veneers are placed. A further complication arises in that the applied stresses are often neither parallel nor perpendicular to the face-grain direction. The most satisfactory method of determining moisture-correction factors for plywood strength properties is to run static tests on a number of *matched* samples of a given plywood at several different values of moisture content. A satisfactory, but somewhat less accurate method, is to assume that the moisture-correction constants for solid wood are directly applicable to plywood. In this case, however, a certain amount of judgment will have to be used in estimating the strength contributed by the veneers running in each of the two directions. (It appears that for all practical purposes the critical buckling strength of plywood panels, in either compression or shear, may be assumed to be unaffected by changes in moisture content.)

Correction for Duration of Load – There is considerable controversy at the present time as to how static-test data should be corrected for the effect of "duration of load." Duration of load can be defined as the time during which the structure fully supports a given maximum load. This is often confused with "loading time," which is something quite different. The latter term can be defined as the length of time required to raise the load from some low value, possibly zero, to its maximum value.

The standard tests from which some of the design wood strength properties were obtained were run at rather slow rates of loading. In order to adjust the test values to the established 3 sec duration of load, a multiplying factor of 1.17 was used. This 1.17 value, however, is a 3-sec *loading-time* factor rather than a *duration-of-load* factor. For this reason it appears that the wood strength properties now used for design purposes may be somewhat too high. Actually, it is incorrect to speak of duration of load unless it is associated with some given loading time, and before a completely rational design wood-strength table can be set up, both of these time factors must first be evaluated from a consideration of actual flight conditions.

A further point which should be noted is that the 3-sec loading-time factor, 1.17 (erroneously called duration factor), has been based entirely upon static-bending tests, and its application to certain of the other wood strength properties is questionable.

In view of the above points, it can be stated that Fig. 3 in the National Advisory Committee for Aeronautics Technical Report No. 344 and the Forest Products Laboratory Mimeograph Report No. 1079 should not be used to correct static-test data for *duration-of-load* effects.

It is realized, nevertheless, that wood structures cannot support loads near the ultimate for long periods of time without suffering some loss in strength. The problem comes up in the strength testing of complete structures and major assemblies of large airplanes in which loading

periods of from 15 to 45 min are necessary to record deflections of the structure at a number of points. At about 85% of the required strength load and higher, every effort should be made to reduce the length of time that the structure must support the full load. At 100% of the required strength load, the structure should not be expected freely to support the entire load for more than about half a minute. It can usually be assumed that any decrease in strength of a complete wood structure or major assembly which might have resulted from a longer duration of load is offset by neglecting to make a moisture-content correction. This is so because most wood aircraft structures are usually found to be below the design value of 15% moisture content.

In static tests of structural elements to establish design allowables, the usual procedure is to apply the load more or less continuously until failure occurs. Some aircraft manufacturers desire to adjust these failure test loads to correspond to some very rapid loading time. In such cases the following points should be kept in mind:

a. A so-called standard or design loading time has not yet been established to correspond to actual flight conditions.

b. Other than for static bending, the effect of various loading times on wood strength and elastic properties has not been determined.

c. If the test load was being increased at the time failure occurred, it is obvious that the duration of load (that is, duration of the maximum load) was zero. Since 3 sec is the established design duration-of-load period, it could be argued that the test strength load should be *decreased* to correspond to this 3-sec duration period.

In view of these points, it is recommended that no attempt be made to correct static-test data of small component parts or structural elements for the effect of either loading time or duration of load.

Summary of Corrections to Static-Test Data for Wood Aircraft

A. Complete structures or major assemblies

1. Specific gravity: No corrections necessary but specific gravity of material in test article should be between $\pm 10\%$ of "average" value given in column 2 of Table 2-1 in ANC Wood Handbook.

2. Moisture content: No corrections necessary. (See item 3 following.)

3. Duration of load: No corrections necessary since the moisture content of the structure will usually be lower than the 15% design value, and this is considered to offset load-duration effects. Structures should not be expected to sustain ultimate design loads for longer than half a minute.

B. Structural elements or small component parts of a structure

1. Specific gravity: a. For solid wood, same as previous item "A-1." For a series of correlated tests, however, scatter in test points will be reduced if all data are corrected, by use of the constants given in Table 2-3 in the ANC Wood Handbook, to the average specific gravity for the species.

b. For plywood, no corrections necessary. Material in test specimens, however, should be representative in both weight and thickness of the plywood to be used in the actual airplane structure.

2. Moisture content: Test data for both solid wood and plywood specimens should be corrected to 15% moisture content by use of the constants given in Table 2-2 in the ANC Wood Handbook. (Correction constants in this table are only approximate for plywood; more accurate values can be determined by tests on matched specimens at several moisture-content values.)

3. Duration of load: No corrections necessary.

■ Detail Design Problems

Plywood Shear Webs for Design of Box-Beams – In the past, the design of box-beam plywood shear webs has been based upon two sources of information, National Advisory Committee for Aeronautics Technical Report No. 344 by G. W. Trayer, and Air Service Information Circular No. 516 by R. A. Miller. Both of these reports were the results of static tests. Because of the differences in test conditions, however, the quantitative design values recommended in each case were not in agreement. In some instances this discrepancy amounted to several hundred per cent, showing the need for more satisfactory design information and clarification.

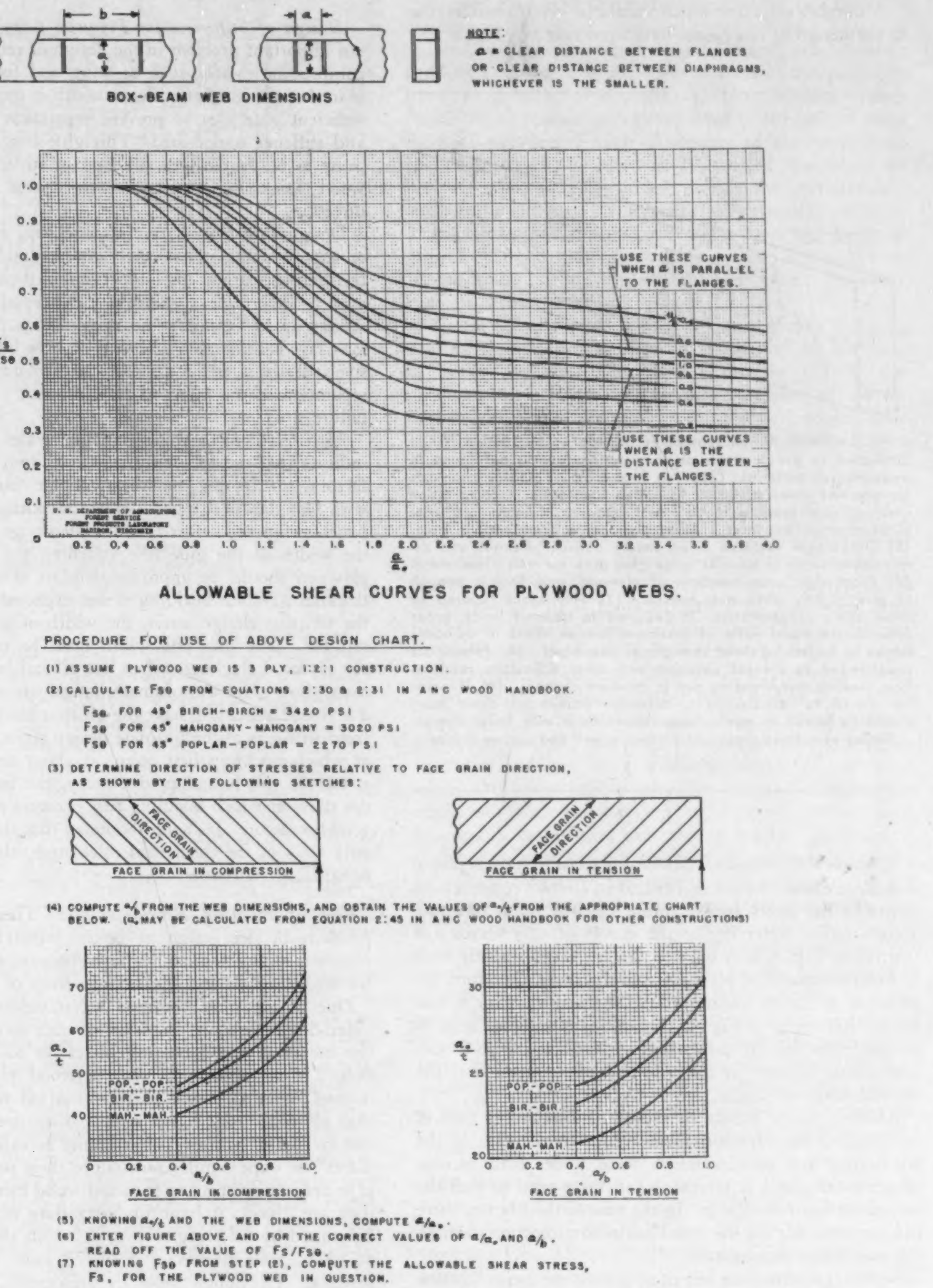
In connection with the current Army-Navy-Civil Research Program, the Forest Products Laboratory in Madison, Wis., made a study of all available test data on the strength of plywood shear webs. This work resulted in the design curve (Fig. 2-19) in the ANC Wood Handbook. Further study and experimental work at the Forest Products Laboratory have produced a revised set of curves, shown in Fig. 5, for the design of plywood shear webs in I-beams and box-beams. The procedure to be followed in the use of this design chart is very simple as can be seen from the step-by-step method which has been outlined just below the curve itself.

Although in a strict interpretation, the curves in Fig. 5 apply only to plywood beam webs, it is believed that they may also be used to calculate the shear strength of other types of plywood shear panels (such as in wing skins or fuselage coverings that have little or no curvature) provided certain precautions are taken. If any edge of a panel is not rigidly restrained against movement in its own plane, the lower set of a/b curves should be used. An example of this might be a plywood panel in the wing covering at the inboard end of an outer panel where the end rib does not afford a rigid spanwise restraint to the edge of the panel.

The shear strength of a panel which is rigidly restrained along all edges in its own plane may be determined by use of the upper set of a/b curves. A panel whose edges are entirely within a larger plywood sheet, or a panel whose edges are restrained on one or more sides by a heavy member and on all other sides by a continuation of the plywood, will fall into this group.

It should be noted that the curves in Fig. 5 are applicable regardless of the direction of the plywood face-grain; the variation in shear strength is compensated for in the terms $F_{s\theta}$ and a_0. $F_{s\theta}$ may be calculated from equations 2:30 and 2:31, and a_0 from equation 2:45 in the ANC Wood Handbook.

Although it is common practice to design metal shear webs in the tension-field range, this same practice is not advisable for an efficient design of plywood webs. It has been found by test that when the face-grain direction is parallel to the tensile stresses (this would be the direction of greatest tensile strength for plywood having equal veneer thicknesses) the buckles become large and the resulting bending stresses reduce the ultimate tensile strength far

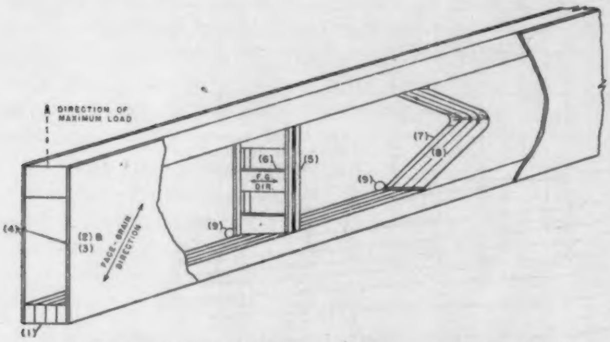


■ Fig. 5 – Procedure for the design of plywood shear webs in I-beams and box-beams

below that normally expected. If the face-grain direction is run perpendicular to the tensile stresses, the resistance to buckling is increased, but in this case the tensile strength of the plywood is low, even in the unbuckled state. Consequently, there does not appear to be any "best way" of running the face-grain direction for plywood tension-field webs. Further, the tension-field buckles tend to project themselves into the glue joints, thus causing premature

failure due to separation of the stiffeners and flanges from the plywood webs.

A number of points which should be given consideration in the design of box-beams have been noted in Fig. 6.



■ Fig. 6 – Detail design feature of box-spar – (1) Tension flange laminated to avoid effects of compression failures not found in inspection of material. (2) Web face-grain 45 deg to spar axis for greatest shear strength. (3) Web face-grain in compression under critical loading for greatest buckling strength. (4) For 50-50 construction, most efficient plywood web material is 3-ply. (5) Diaphragm verticals fitted snugly between flanges and dimensioned so as to provide large glue area for web attachment. (6) Diaphragm cross-members of plywood, and flexible enough to give slightly when web buckles. (7) Filler-block tapered to avoid stress concentration. If designed to transmit loads, taper ratio should equal ratio of maximum tension stress in adjacent flange to horizontal shear strength of filler-block. (8) Filler-block crossbanded to prevent checking and other difficulties resulting from swelling and shrinking due to moisture changes. (9) If water has access to interior, ¼-in. diameter (minimum) drain holes should be made in each compartment flush with lower flange. Similar vent holes should be placed in web near upper flange

Efficiency of I-Beams – Box-beams seem to have replaced to a large extent the use of I-beams in aircraft construction, although the latter type is more efficient on a strength-weight basis. Referring to the curves of $a_{0/t}$ versus a/b just below Fig. 5, it is obvious that a_0 varies directly with t. For the same total plywood web thickness, therefore, the value of a_0 for an I-beam would be twice that for a box-beam. Referring to Fig. 5, this means a value of a/a_0 of one-half that for the corresponding box-beam with a very appreciable increase in the allowable shear stress over the normal range of design.

Another factor which should be considered is that of buckling of the plywood shear webs. In the case of the box-beams, any buckles which form under load become deeper as the load is increased and these tend to pull the web away from the flange. In the case of the I-beam, there is no opportunity for the web buckles to project on through the web-flange attachment.

Some arguments may be raised against the use of I-beams because of the difficulty in attaching ribs. Such arguments have very little basis as it is quite simple to use the web stiffeners as rib-attachment points, or it is also possible to devise various "mechanical shear joints" which take advantage of the I-beam cross-section profile.

This discussion indicates that designers would do well to consider the use of I-beams for plywood-covered wings and not follow along blindly in the use of the more or less standard box-beams.

Design of Stiffeners for Plywood Compression Panels – An important problem in the design of reinforced plywood compression panels, such as wing and fuselage coverings, is to determine what stiffener width is required to provide sufficient glue area to prevent separation of the plywood and stiffener under load. This glue-area requirement appears to be primarily a function of stiffener spacing, plywood thickness, plywood construction, and face-grain direction.

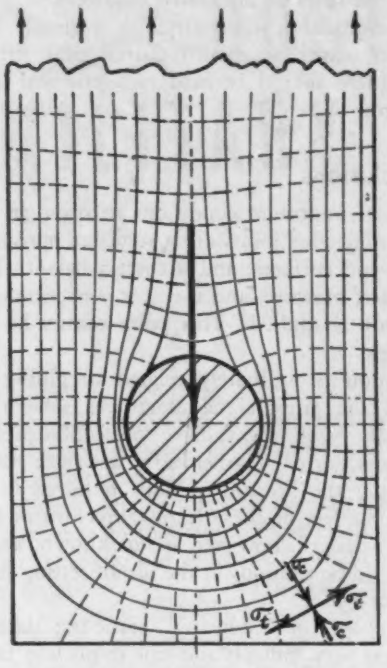
Design information on this problem, either theoretical or experimental, is extremely meager and aircraft manufacturers are urged to conduct stiffened-panel compression tests to determine a suitable stiffener width rather than arbitrarily to assume some value. Such preliminary tests may save considerable time and expense by avoiding extensive redesign if, at the time the completed airplane is tested, separation of the plywood and stiffeners causes premature failure.

Such test data as were available at the time of writing indicate that the following rule of thumb should be satisfactory for 45-deg mahogany-poplar multiple plywood with Douglas-fir stringers. When buckling of the plywood panel is expected before the ultimate design load is reached, the width of the glue line attaching the stringer to the plywood should be approximately six times the plywood thickness. When buckling is not expected to occur below the ultimate design stress, the width of glue line may be reduced. The glue-line width need be only three times the plywood thickness when the critical buckling stress is more than twice the ultimate design stress. Theoretically, a plywood panel will have the greatest tendency to separate from stringers at the ultimate design stress when the stress at which buckling first occurs is about two-thirds of this stress. In the foregoing discussion, the buckling stress is the stress at which buckling first becomes noticeable in the actual structure. It should be noted that this stress may be only 50% of the theoretical calculated value, due possibly to initial eccentricities.

Use of Plywood Bearing Plates – There is a serious problem in the design of bolted joints for solid wood members under tension loadings because of the tendency for wood to split and shear-out in front of the load.

One of the most common recommendations for good detail-design practice in wood aircraft construction is for the use of plywood bearing plates for all bolted connections. It is unfortunate that plywood plates have been termed "bearing plates" as this has led many to believe that plywood itself has a high-bearing strength, which is not so. Actually, these plates should be called "reinforcing plates" or some similar term since their primary function is to prevent splitting of the solid wood members to which they are glued. A basic understanding of this point can be gained by referring to Fig. 7, which shows the results of photoelastic studies of a bolted joint in an isotropic material. (Although wood is anisotropic, it is believed that this figure is at least qualitatively correct when applied to a wood member.) The solid lines represent the lines of principal tensile stress while the dotted lines represent the companion principal stress.

Since wood is very weak in tension perpendicular to the grain, it is obvious that the tensile stresses shown in Fig. 7 will cause the member to split open at very low loads. If



• Fig. 7 – Principal stresses set up in isotropic plate by a bolt load

however, a piece of plywood is glued to each bearing face of a bolted member, the plywood will resist the stresses tending to cause cleavage, thus enabling the solid wood to develop a much greater shear strength before failure.

It can also be seen from the stress trajectories in Fig. 7 that the direction of maximum shear stress just below the bolt is essentially parallel to the direction of loading. For this reason plywood reinforcing plates should be laid so that the face-grain direction runs 45 deg to the direction of loading.

For the same overall thickness it has been found by test that a plywood-faced bolted member may develop as much as 50% more strength than will the same member without plates.

The thickness of plywood to be used is necessarily a function of the thickness of the bolted wood member. It is recommended that a minimum thickness of ⅛-in. 45-deg plywood be used on each face for a total thickness up to 2 in. and varying linearly thereafter to ¼ in. for a total thickness of 5 in. The plywood should be of 50-50 construction.

It is further recommended that mahogany-poplar plywood be used for reinforcing low-strength woods, such as spruce and poplar, and that birch-birch plywood be used for reinforcing high-strength woods such as birch, maple, hickory, and walnut. Composite members made in this manner will give bearing strengths almost as high as would be obtained from the use of "impreg" or "compreg."

Use of High- and Low-Density Woods in Combination – Although it is entirely possible to use high- and low-density woods in combination in highly stressed members satisfactorily, the differences in elastic properties are a potential source of trouble. In certain aircraft structures, particularly wings, premature failures in static tests have sometimes been caused by the elastic incompatibility of high- and low-density woods.

A specific example of an "incompatibility" failure is one which occurred on a wing with hickory spar caps and square-laid poplar plywood covering (face-grain direction running spanwise). In the stress analysis of this particular structure the skin was assumed to carry no bending stresses. Nevertheless, tension failures occurred in the skin at about 80% of the design load. A review of the stress-strain characteristics of the two woods revealed that while the "E" of poplar was much less (hence the corresponding stress for a unit strain was less) the total work which the hickory was capable of doing under load was so much greater that the ultimate strength of the poplar was exceeded while the hickory was yet well within the elastic range.

The remedy for such an occurrence is merely to reduce the "E" of the plywood covering by rotating the face-grain to be 45 deg with the spanwise direction. This latter arrangement is particularly desirable in view of the fact that the original assumptions in the above-mentioned stress analysis are more nearly fulfilled and the torsional properties of the wing are improved.

Notes on Design of Leading-Edge Structures – An interesting case of leading-edge failure on a conventional two-spar, trussed-rib, fabric-covered wing (plywood nose) was observed during static tests. In the design of this particular wing, the plywood leading-edge cover had been disregarded structurally, even though it was securely fastened to the ribs and the front spar. As the wing deflected under load, the leading-edge plywood cover was stressed accordingly. Failures began to occur at as low as 40% of the design load in the plywood nose cover in the vicinity of access cut-outs, and finally the entire leading-edge structure was torn from the wing at less than 80% of the design load. Designers believe they are being conservative by neglecting the load-carrying capacity of certain parts, whereas in reality they are inviting trouble. A failure or series of failures, caused by structural members which are arbitrarily considered nonstructural, is just as costly and even more irksome than failures in members which were given sufficient consideration in design.

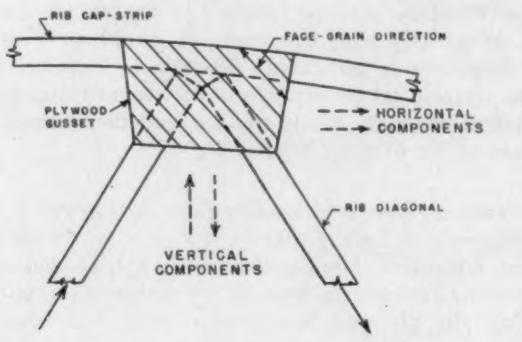
A general weakness of leading edges in stressed-skin wood structures appears to be the tension attachment of ribs to spars. It is fairly easy, in the usual case, to obtain adequate shear connections with corner blocks or plywood angles, but tension connections are affected by skin stresses from sources other than the direct rib loads. This fact, coupled with the inherent difficulties involved in making a satisfactory tension connection in wood structures has caused considerable "leading-edge trouble."

Plywood Gussets for Trussed-Type Ribs – Little or no attention is given to the design of plywood gussets for use in attaching diagonals to cap-strips in the conventional truss-type rib. This oversight usually results in premature failures in rib static tests and consequent delay while the "gusset trouble" is remedied. The most efficient type of plywood gusset for a particular location will depend upon the relative magnitudes of the net horizontal and vertical forces acting on the gusset.

In general, it can be stated that the plywood gusset should be of 50-50 construction of not less than three plies. For ease in gluing, the density of the outer plies of the

gusset and the density of the cap-strip and diagonal material should be approximately the same.

If the greatest strength is desired in the vertical direction and the glue area between the gusset and cap-strip is critical, the plywood face-grain direction should be run parallel to the rib cap-strip. (This assumes 50-50 construction is being used.) If the greatest strength is desired in horizontal shear, such as when the horizontal forces in the diagonals are additive, the plywood gusset should be laid so that its face-grain makes an angle of 45 deg with the cap-strip. A further point to be noted in the latter case is that the plywood face-grain should run in the general direction of the tension load. (See Fig. 8.)



■ Fig. 8 – Design features of typical trussed-rib plywood gusset – (1) Plywood gusset should be of 50-50 construction and not less than three plies. (2) Outer plies of gusset and cap-strip material should be of about the same density. (3) Plywood face-grain should be 45 deg to cap-strip, and run in the direction of the tension load

Avoidance of Stress Concentrations – The proportional limit of wood in tension is very near its ultimate strength. This, combined with the weakness of wood in shear and cleavage, makes it *essential* that abrupt changes in section, re-entrant cuts, odd-shaped lightening holes, and the like be avoided wherever possible. Unfortunately, stress concentrations in wood cannot relieve themselves by plastic yielding as they do in metals.

Reinforcing blocks should be tapered in thickness or in width or both, so that they will pick up stress as uniformly as possible. When fish-mouthed filler-blocks are used in box-spars for stress transfer, the taper ratio of the fish-mouth should be determined by the ratio of the maximum tension stress existing in the adjacent flange to the horizontal shear strength of the filler-block. (See Fig. 6.)

Replacing Solid Glue-Blocks with Plywood Angles – A few tests which have been made indicate that plywood angles (bent up from 45-deg plywood) will carry safely 250 psi in shear. Since their weight is only one-third to one-half that of an equivalent solid glue-block, some weight saving can be realized from their use.

Reduction in Fitting Size and Number of Fittings – A staggered bolt pattern which will give a smaller and more efficient fitting can be used if suitable plywood plates are glued to the faces of the bolted members.

The weight of a wood airplane is greatly increased by the use of extra material required near fittings; consequently, these should be held to a practical minimum in any design.

■ Conclusions

The most important conditions involved in the production of strong glue joints are a sufficient spread of glue on smooth wood surfaces, and a correct balance between the temperature, pressure, and the glue consistency at the time the pressure is applied. This point cannot be emphasized too strongly.

Fabrication is intimately related to gluing operations, and until the aircraft manufacturers solve their gluing problems they will always have fabrication problems. The use of above-normal temperatures has been recommended for reducing the pressing periods required for cold-press glues. The effect of these high temperatures on the final glue-joint strength, however, is not known, and manufacturers should be cautious in the use of setting temperatures in excess of 150 F.

The problem of "reducing" static-test data for wood structures is very difficult and can in no way be associated with the more-or-less standard correction procedures that have already been established for metal structures. Certain factors which affect the strength of wood tend to offset one another, and corrections to static-test data for complete wood structures are unnecessary, provided some control is maintained over the specific gravity of the material in the test article.

The somewhat poor reputation that wood aircraft structures have acquired in the past year in static tests can be attributed to a lack of knowledge of plywood strength and elastic properties, as well as to poor detail design practices. It should be remembered, however, that both of these items will be overcome as a background of design information and experience is gained similar to that already available for metal structures. What constitutes poor detail design practice in any type of structure may not be realized until after a costly static-test failure or service failure is encountered. Such failures provide the foundation for more efficient designs in the future.

■ List of References

1. "ANC Handbook on the Design of Wood Aircraft Structures," July, 1942. (Restricted)

2. NACA Technical Report No. 354, 1930 (also Forest Products Laboratory Mimeograph R-1079), "Aircraft Woods: Their Properties, Selection, and Characteristics," by L. J. Markwardt.

3. *Product Engineering*, Vol. 10, December, 1939, pp. 548-551: "Photoelastic Analysis of Stress Distribution in Pin and Plate Joint," by A. G. Solakian.

4. U. S. Department of Agriculture Technical Bulletin No. 479, 1935, "Strength and Related Properties of Woods Grown in the United States," by L. J. Markwardt and T. R. C. Wilson.

5. U. S. Department of Agriculture Technical Bulletin No. 282, 1932, "Strength-Moisture Relations for Wood," by T. R. C. Wilson.

6. "Wood Aircraft Fabrication Manual," July, 1942 (Restricted)

DUST PROBLEMS in MILITARY VEHICLE OPERATION

by L. F. OVERHOLT

International Harvester Co., Inc.

ON April 5, 1943, a letter was written by Brig.-Gen. John K. Christmas, stressing the seriousness of the dust problem in military vehicles. He stated, "Better protection against dust is one of the most pressing needs for the successful operation of mechanized war equipment. Dust is as real an enemy to our mechanized war equipment as the enemy themselves and it will require the combined efforts of all companies to defeat it."

The tractor industry has long recognized dust as a real enemy, and for nearly 30 years has been combating this problem. The dust problem, however, has stubbornly resisted a universal solution, as each new experience usually demanded a new standard. For instance, the early tractors, used chiefly for plowing and threshing, were equipped with open clutches, exposed final-drive gears and, in many instances, no air cleaner was supplied. Eventually, as the field of operations was extended to cultivation of crops and building and maintenance of roads, it became necessary to enclose the moving parts of the tractor and to protect engines against the entrance of dust.

The air cleaner was one of the first requirements for tractor engines and has probably received more attention than any other accessory. The first models were the dry type, which extracted the dust from the air centrifugally and were capable of removing small pebbles and larger dust particles with an efficiency anywhere from 50 to 75%. Engine life with these cleaners was short – 100 to 150 hr in severe dust conditions in some parts of the country.

Air cleaner development ran the gauntlet of the dry centrifugal type, the water-bath cleaner, the oil-wetted filter, and finally the currently used oil-bath cleaner.

Examination of data on various laboratory air cleaner efficiency tests is proof that the average oil-bath air cleaner in commercial use on tractors is very efficient; practically 100% in extracting dust particles larger than 10 microns from the air stream entering the engine.

Dust particles smaller than 10 microns constitute a serious wear problem in some sections of the country and are more difficult to trap, but a well-designed cleaner will remove from 80 to 95% of these finer particles for an over-all cleaner efficiency of 97 to 98% by weight.

A series of tests was recently made to determine particle size of dust escaping through two air cleaners having different over-all efficiencies. One cleaner, using an extra scrubber element to bring about a more intimate mixing of the dust and oil bath, had an over-all efficiency of 98.5%. The other cleaner, without an extra scrubber element, had an efficiency of only 95.5%. Microscopic examination of absolute cleaner cloths, used in series with the air cleaners, failed to reveal any dust particles larger than 5 microns. This was particularly interesting as it demonstrated that although a large portion of fine dust is removed by impingement on oil-flushed surfaces, a considerable portion still will pass through the cleaner unless a more positive means is provided to bring about the emulsion of dust with

AS an approach to immediate and greater effort toward better protection of our military vehicles against dust, Mr. Overholt reviews the progress made in a 30-year war by the tractor industry to reduce this menace. Protection of the engine and vital parts of the chassis is a major problem.

Efficient air cleaners are available but lack of space hampers proper installations on mechanized war equipment. Leakproof connections between air cleaners and engines are as essential as efficient air cleaners.

Fine dust particles cause serious engine wear when large quantities enter the engine. Intense dust concentrations necessitate too frequent servicing of air cleaners when tanks and combat vehicles operate on deserts.

All vital parts of mechanized war equipment must be protected against dust. Half-way measures cannot be tolerated. Rapid wear of some parts can be reduced but not stopped, necessitating simple constructions to provide an easier service problem. Many dust problems in mechanized war equipment are new and new standards will be required.

★ ★ ★

THE AUTHOR: Lloyd F. Overholt (M '39) is assistant to the chief engineer in charge of engine and tractor development of the Gas Power Engineering Department of International Harvester Co. Long interested in the particular problem of protecting mechanical equipment against abrasion by dust, he serves on the SAE War Engineering Board Dust Technical Committee. Mr. Overholt's concern with farm and industrial tractors and industrial engines dates from student days in mechanical and agricultural engineering at the University of Minnesota. His interest in the field took him in 1916 to a position as development and experimental engineer with Minneapolis Steel and Machinery Co. He shifted to International Harvester in 1931.

[This paper was presented at the SAE War Matériel Meeting, Detroit, Mich., June 10, 1943.]

oil. The dust used during these tests was prepared from soil obtained near Phoenix, Ariz., by the A.C. Spark Plug Division for testing Ordnance air cleaners.

Photomicrographs of this dust show the 200-mesh dust before it was fed into the air cleaner (Fig. 1) and after it had passed through the air cleaner and was caught on the absolute cleaner cloth (Fig. 2). The particles in Fig. 2 were coated with a very thin oil film as they passed through the air cleaner and appear as agglomerates in the picture. In Fig. 3 the particles shown in Fig. 2 were treated with a solvent to remove the oil film and segregate the particles. Actual particle size is indicated by the scale across the picture. (Scale = 2.5 microns per division. One micron = 0.00003937 in.)

Particle size analysis of this dust was given as:

Microns	%
0-5	39
5-10	18
10-20	16
20-40	18
over 40	9

We do not mean to imply by these statements that all oil-bath cleaners will always collect all dust particles down to 5 microns, because some dusts do not emulsify readily with oil, and therefore particles as large as 15 or 20 microns might pass through.

Through the courtesy of C. T. O'Harrow of the Allis-Chalmers Mfg. Co., we quote data obtained during tests made at the Allis-Chalmers laboratory to determine effect of dust particle size on rate of engine wear: "In order to get information in this regard, we have obtained dusts from three localities, namely, Phoenix, Ariz.; St. Anthony, Idaho; and West Allis, Wis. The dust particles of these various dusts, which fall into the classes of 0 to 5 microns, 5 to 15 microns, and 15 to 30 microns, have been separated and these have been fed into engines for observation of wear produced."

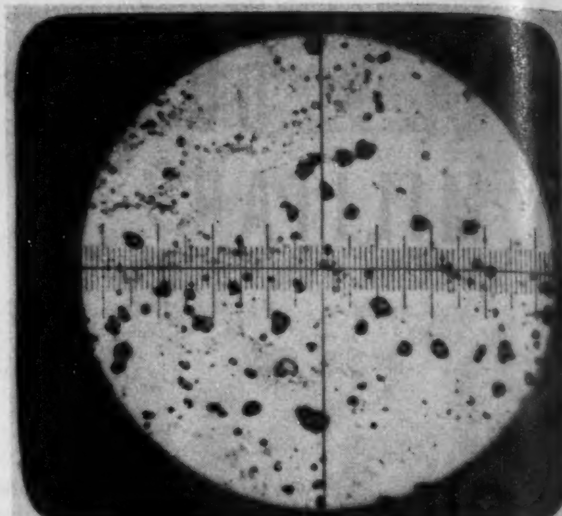
Mr. O'Harrow, in summarizing his data, states: "These data show that regardless of origin there is a distinctly different rate of wear produced by fractions in the various size ranges.

"As a generalization it can be said that a 15- to 30-micron dust produced approximately 2½ times as much wear as the 5- to 15-micron dust and the 5- to 15-micron dust produced approximately 1½ times as much wear as the 0- to 5-micron dust.

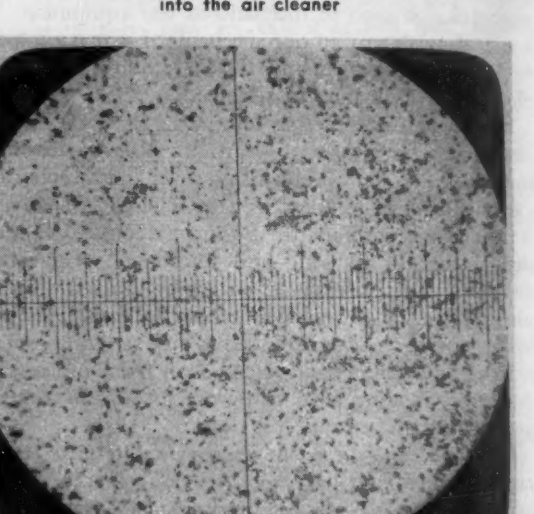
"There is some difference in the wear produced by the dusts of various origins. The dust collected in the Allis-Chalmers field at West Allis produced the highest rate of wear, while the dust gathered at Phoenix, Ariz., produced the lowest rate of wear.

Mr. O'Harrow's findings show that even the very fine dusts will produce engine wear, but more in some parts of the country than in others. Abrasiveness of dust is to a large extent dependent upon the percentage of quartz present and to the sharpness of the particles. Samples of dust from Phoenix show between 55 and 65% pure quartz, but many of the particles appear smooth and nearly round under the microscope, where they normally occur as prisms or hexagonal crystals. Abrasion of particles by the wind blowing them across the desert has probably reduced the particles somewhat, but regardless of form or size these dust particles will produce engine wear if large quantities enter the engine.

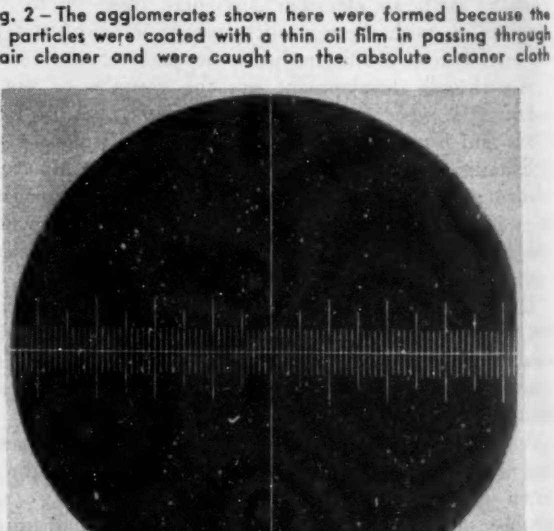
Application of the air cleaner is equally as important as



■ Fig. 1 – 200-mesh dust from Phoenix, Ariz., before it was fed into the air cleaner



■ Fig. 2 – The agglomerates shown here were formed because the dust particles were coated with a thin oil film in passing through the air cleaner and were caught on the absolute cleaner cloth



■ Fig. 3 – After the particles shown in Fig. 2 had been treated with a solvent to remove the oil film and segregate the particles

having an efficient air cleaner. Experience has shown the air cleaner must be so located as to be readily accessible for servicing, and all connections between the air cleaner and the engine must be simple, reliable, and convenient for frequent inspection for leaks.

Servicing of the air cleaner is likewise important. It was necessary for tractor companies to resort to extensive educational campaigns before satisfactory engine protection was accomplished. These campaigns not only stressed the care and maintenance of the engine air cleaner and induction system, but also engine crankcase breathers and oil level gages, both of which are likely points of dust entry if not properly protected.

Paralleling measures taken to protect the engine, similar precautions were developed to protect the vital parts of the power train and running gear of the tractor, as well as the electrical equipment and instruments. This involved the development of dust seals, special enclosures, and shields.

Motorizing the army is a new experience. It has produced many new types of vehicles and, as was the case with the tractor, each tank, combat, or transport type of vehicle will have to be analyzed separately and the susceptibility to dust of each treated individually.

In military operations, tanks and combat vehicles are driven in echelon or column formation across the desert, where dust flows like water after the first vehicles have broken the crust. Dust concentrations of 0.17 g per cu ft have been recorded near the ground and 0.035 to 0.055 g at 6 to 8 ft above the ground; 0.045 g of dust per cu ft gives zero visibility.

Tractor users have learned that groups of tractors required to operate in dust concentrations greater than 0.025 g, except for very brief intervals, should separate to keep out of each other's dust cloud. Combat vehicles, however, operate as necessity demands and, therefore, must be dustproofed accordingly, which means they must be protected against the severest rather than the average dust conditions.

All parts of the machines were affected in the African desert campaign, but one of the chief complaints was too frequent service required for the engine air cleaners. Complete data concerning the servicing of air cleaners under actual combat conditions on the desert will probably not become available until after the war is won. However, factual information obtained from manufacturers' field men and Ordnance representatives who have had experience with combat vehicles on the desert is proof that the air cleaner service interval is much too short. This applies to all types of motorized equipment, and by way of illustrating the quantity of dirt to be removed by the cleaner a 500-hp tank engine, operated continuously under a 50% load factor in a dust concentration of 0.045 g per cu ft, would take in about 4.9 g of dust per hp-hr. On the basis of a 10-hr operation 12.25 kg, or roughly 27 lb, of dust would be collected. This compares favorably with published data on conditions encountered in North Africa, and this amount of dust would require a single air cleaner 24 in. in diameter or two 15-in. cleaners for a desired 10-hr service interval.

Air cleaners larger in diameter than 12 in. are awkward to service and, when too large, are impractical. Military vehicles must be built as compact as possible: space for accessories is always at a premium, and air cleaner diameter must be kept at a minimum without restricting the engine induction system. These same space limitations frequently prevent the air cleaner from being installed in the most advantageous location with regard to dust concentration and ease of servicing.

Position of the air cleaner and particularly the air cleaner inlet should always be in the zone of least dust concentration about the machine, as improperly located cleaner inlets have necessitated servicing every 2, 3 or 4 hr; a 12-in. diameter cleaner holds over a gallon of oil, and too frequent servicing consumes huge quantities of oil, where an armored division is involved.

Excessive engine wear resulting from leaky air cleaner connections was another source of complaint; this may have been due to mediocre installations and/or inexperienced mechanics. Chaff, lint, and leaves from vegetation also caused trouble by clogging air cleaner filters.

These complaints are typical of tractor experience. Screens over the air inlets will prevent vegetation entering the air cleaners and improved installations will overcome dust leakage into the connections, but extending the service interval of the air cleaner to, say, 10 hr of operation offers a more serious problem.

Extension pipes to raise the air cleaner inlet above the heavy dust cloud have been provided for tractors, and these have extended the time between service intervals considerably. Vertical extension pipes for tanks and combat vehicles would not be practical, however, as they would interfere with other equipment. Auxiliary cleaners of the dry centrifugal type located ahead of the oil-bath cleaner, to remove approximately 50% of the dust from the air before it reaches the oil-bath cleaner, seem to offer a solution. There is some objection to the use of these cleaners, however, as they tend to reduce the over-all efficiency of the oil-bath cleaner, but well-developed combinations should not lower the efficiency more than 0.5%, which appears to be a low price to pay for a 100% increase in air cleaner service periods and a 50% saving in oil for the air cleaner. On the other hand, we should not lose sight of the fact that the best air cleaners have an over-all efficiency of about 98%, and the 0.5% loss in efficiency would represent an increase of 25% in the amount of dirt that would enter the engine.

The chief problems in dustproofing the engines for mechanized war equipment, as we have attempted to emphasize, are:

1. To provide an accessible location for the air cleaner where it will not be subjected to high temperatures and can be serviced without the use of tools.
2. To locate the air cleaner inlet in a zone of low dust concentration as a means of increasing the service interval of the cleaner.
3. To save oil and service time.
4. To provide simple but durable connections between the engine and air cleaner.
5. To provide a filter for the engine crankcase ventilating system.
6. To seal the oil level bayonet gage.

Dust protection for the vital parts of the power train, running gear, electrical equipment, and controls, as for the engine, has offered many difficulties.

Clutches for the half tracks, scout cars, and transport vehicles should not cause much trouble, if all leaks are sealed and the clutch is completely enclosed and the housing ventilated through an air filter which can be easily cleaned. Tank clutches, however, cannot be as completely enclosed as in lighter vehicles. This is due partly to the

fact that some engines adapted to tank use are not designed to accommodate a complete clutch enclosure and partly to heat accumulation in the clutch. Dust getting into partially enclosed clutches will soon cause them to stick in the fully or partially engaged positions, and then gears cannot be shifted or clutches will slip and burn out.

Deflector shields are of little value. The only immediate solution seems to be in the elimination of the clutch housing and to depend upon the wind created by the clutch and centrifugal force to throw the dust out. This is not conducive to long clutch life, but it has extended the useful life of some clutches several hundred per cent. The ultimate solution should be a completely enclosed dustproof clutch with a water jacket or filtered air blast for cooling.

Clutch throwout bearings for open clutches must be the fully enclosed, permanently lubricated type; otherwise they will fail in a very few hours.

Transmissions and final drives have not caused as much trouble as other units because they are sealed against oil leakage; however, the seals must be tight and the shafts kept dry at the seals; otherwise dust will lodge in the oil and grind into the seals and ruin them. Breathers for these units must be of the filtered-air type and easily accessible for cleaning.

Dust entrance into electrical equipment, including starting motors, generators, switches, and distributors, has caused many failures. Distributors have failed in two or three miles due to dust glazing over the contact points. This is an old story to the tractor industry, which found it necessary to enclose magnetos completely, even to the drive shafts, to prevent dust entrance; any necessary vent holes were protected with air filters. Battery distributors provided a similar problem and required similar treatment. Vent holes in distributor caps, and joints between caps and bodies of distributors, had to be sealed. This practice likewise is necessary for war machines.

Generators fail rapidly in heavy dust concentrations because of bearing wear letting the armature rub on the pole faces. Commutators and brushes wear rapidly. Completely enclosing the small low-output generators has been successfully accomplished in tractor service, but in the case of larger, higher-output units, heat dissipation from the generator is not rapid enough to maintain the generator temperature below the danger point, and complete enclosures cannot be used until satisfactory ventilating systems are worked out. This indicates larger generators must be left open to allow dust to pass through and not lodge in the generator frames. Care must be exercised in this type of installation to shield the generator from oil because dust sticking in the oil would soon pack the generator. Permanently lubricated, fully sealed bearings should be used with the open generator.

Electric switches for Ordnance equipment require complete sealing to avoid rapid wear and open contacts by dust. In tractor service, poorly sealed lighting switches load up with dust in a very short time and the electrical controls are open-circuited. Complete gasket enclosure has been a satisfactory solution. Starting-motor switches, at times, have such a coating of dust driven into the copper contacts that no amount of foot pressure will ensure electrical contact. Military requirements are very severe, and enclosing switches to the point where they are watertight will be the only satisfactory way to solve this problem.

Instruments fail due to dust filtering into the vital parts. Tractor experience has proved that early failure can be expected unless these units are watertight. Specifications set up in the International Harvester Laboratory require immersing in water under 5 psi pressure for 5 min with no leakage.

Controls are a source of trouble due to dust causing them to bind and wear rapidly. Flexible-wire types are bad in this respect unless both ends of the guide tube are sealed with boots. Guide tubes made of coiled wire permit dust to sift through unless taped for their entire length. Formed tubular guides are more satisfactory.

Dust in wheel brakes is not a serious problem where the brakes can be left open so the dust can get out, but mud then becomes a very serious problem; in fact such a problem that everyone is about ready to give it up. Dirtproofing of brakes offers a real development problem. Perhaps a lesson can be taken from the tractor industry, which has provided housings inside the tractor frame for brakes or used enclosed brakes mounted on a transmission shaft.

Air cylinders for brakes of both the high-pressure and vacuum types should be equipped with efficient air-vent filters. These filters should be of good size and easily accessible for cleaning. Small filters clog and cause the piston to remain at the outer end of the stroke due to vacuum being created below the piston in the pressure type. Dust leaking through the vacuum type passes into the inlet manifold and causes engine wear.

Cooling systems for tanks have offered many problems. In water-cooled installations *lack of space* has limited the frontal area of radiators. Deeper cores having closely spaced cooling fins are required and these clog easily, particularly when oil and moisture are present to catch the dust and chaff. Cleaning of these cores is next to impossible because the foreign matter tends to pack in and bake onto the surfaces. Aircooled units likewise can be clogged by chaff and dust if any oil is present. Every precaution should be taken to stop all oil leaks and to remove any trace of oil from the surfaces. Close attention should be given in designing this equipment to avoid constructions that will cause eddy currents in the air stream or allow pockets in which dust can lodge.

Fan belt life is short in heavy dust concentrations. Fan belts have always been a source of annoyance to the designer and service man because dust drastically reduces the normal life of belts and belt pulleys. Tractor experience has proved that fan belts should be run high in the pulley grooves and loaded as lightly as possible to keep belt and pulley wear to a minimum. For operation in severe dust conditions it would be advisable to use two belts where only one would normally be used.

We have attempted in a general way to outline the dust problem in mechanized war equipment but we have not attempted to cover the entire problem. In the tractor industry experience has proved that all vital parts of the machine must be protected against clogging or abrasion by dirt. Lessons obtained the hard way have taught designing and development engineers that the dust problem is not a stepchild to be pushed around but a real enemy that cannot be defeated by half-way measures. Continued effort over a period of nearly 30 years has reduced wear in tractors caused by dust to an acceptable minimum. Commercial and pleasure vehicles likewise have been protected as conditions required but these conditions were never very severe. Mechanized war equipment offers many new problems, but with the ingenuity of the American engineer these will be solved.

SPECIAL ADDITION AGENT STEELS

by R. B. SCHENCK
Chief Metallurgist,
Buick Motor Division, General Motors Corp.

THE term "special addition agent" refers to a group of ferroalloys containing boron, which have the property of markedly increasing the hardenability of many steels when added in relatively small amounts. These alloys are also known as "needling agents" or "intensifiers."

The main reason for the interest shown in these materials at this time is the promise they offer of conserving critical alloying elements. It now seems quite certain that, by suitable adjustment of steel chemistry, these alloys can be used to replace important amounts of such elements as nickel, chromium, and molybdenum.

CERTAIN ferroalloys containing boron, known as "special addition agents," possess the property of markedly increasing the hardenability of many steels when added in relatively small quantities.

These additives offer promise of conserving critical alloying elements by their ability to replace important amounts of nickel, chromium, and molybdenum.

The additive treatment of steel from a commercial viewpoint is relatively new, having started in 1938.

In general, it may be stated that, with respect to hardenability and mechanical properties, a carbon steel can be made equivalent to a low-alloy steel and a low-alloy steel equivalent to a high-alloy steel by additive treatment.

The amount of additive required varies, depending upon the type of additive, the composition of the steel, and the degree of deoxidation. Uniform melting practice is essential to good results.

In spite of the exacting requirements to be met in melting treated steels, a number of mills have demonstrated their ability to melt heat after heat with as good uniformity as untreated steels. This has been borne out by the experience of the author's company and other users with a large number of heats including a number of different compositions.

Tests data pertaining to treated steels have been collected and published in an AISI report. Another survey for additional data is being planned.

A cooperative test program involving nine heats of treated steel is now in progress and a second program is about to be inaugurated.

Buick laboratory tests described herein show the effect of varying amounts of additives and comparative properties of a number of treated and untreated steels.

■ ■ ■

The Author: ROBERT B. SCHENCK (M '22) has been chief metallurgist of Buick Motors Division, General Motors Corp., for a quarter of a century, and is close to the extraordinary metallurgical developments of those years. His career has been devoted to steels: with Carnegie Steel at Homestead, and with the Western Mott Co., prior to joining Buick. A native of Beacon, N. Y., Mr. Schenck was graduated from Lehigh University in 1909.

■ Historical

Probably what was the first attempt in this country to use boron as an alloying element in steel occurred shortly before World War I. In fact, this author's college thesis was entitled "The Manufacture of Ferroboron from Colemanite." Colemanite is a calcium borate. Little came of this work and interest soon died, which is not surprising when we consider the state of metallurgical knowledge at that time, especially with respect to hardenability.

In the early 20's there seems to have been some activity relative to boron steels in Germany. In 1921 an application was filed and in 1924, a U. S. patent was issued to Richard Walter of Düsseldorf pertaining to the use of boron in steel. Apparently nothing developed from this of commercial value and boron steels again lay dormant until 1938.

A number of special deoxidizers containing various combinations of such elements as aluminum, silicon, titanium, zirconium, vanadium, and calcium were being exploited at that time. One of these, which it was found later contained about 0.20% boron, was added in the ladle to a heat of basic open hearth GM 1340-A steel for Buick Mo-

[This paper was presented at the SAE National Tractor Meeting, Milwaukee, Wis., Sept. 24, 1943.]

tor Division, General Motors Corp. Routine tests conducted by Buick on that type of steel included Jominy hardenability and tensile properties on oversize 0.505-in. diameter test bars oil quenched and drawn at 450 F. The test results obtained on the special heat were so unusual that immediate steps were taken to verify them. Additional heats, treated in the same manner with ladle additions and others with ingot mold additions, confirmed the original results. Since that time a great deal of experimental work has been done by this company, by other users, and by steel producers, new addition agents have been developed, and thousands of tons of special addition agent steels have been made and used for a great variety of applications.

Effect of Special Addition Agents

The general effect of the special addition agents is to increase the hardenability and improve the mechanical properties of steel in the quenched and drawn state. In other than the quenched and drawn state, the additive seems to lower the mechanical properties. Although there may be exceptions, this tendency has been noted in a number of instances.

The improvement in mechanical properties resulting from an additive treatment is confined mostly to the plain carbon and low-alloy steels, and this improvement is most pronounced at low draw temperatures in the range of 300 to 500 F. In sections which permit through hardening, treated SAE 1000 and 1300 steels will develop tensile and Izod values of the same order as those of the high-alloy steels.

In general, it may be stated that, with respect to hardenability and mechanical properties, a special addition agent treatment can make a carbon steel equivalent to a low-alloy steel and a low-alloy steel equivalent to a high-alloy steel.

Use of Special Addition Agents

The standard practice is to make the special addition in the ladle after all other additions have been made. The amount of addition agent to be used depends upon its composition and upon the composition and degree of deoxidation of the steel. Conditions may even be such in a poorly deoxidized heat that the effect of the addition agent is completely lost. If increasing additions of a given type of additive are made to different ingot molds of the same heat, the hardenability will be found to increase with the amount of additive up to a maximum and then decrease with further increases of additive. This is well illustrated in Figs. 3A and 3B. Maximum properties usually coincide with maximum hardenability, and the amount of addition agent required to produce these maximum values is generally considered the optimum. Smaller additions may be extremely beneficial but there is some doubt as to their uniformity of action. Larger additions than the optimum result in no increased benefit and show a tendency to lower the mechanical properties. The fact that these additives react effectively when added in the ingot mold has been of great value in development work. If this were not true, it is probable that not nearly as much information as we now have would be available.

From the above, it is evident that uniform melting practice in each mill is of prime importance in the use of addition agents. One mill may require an addition of 4 lb per ton of a certain additive while another mill may find 2 lb per ton sufficient. Each mill should determine the optimum addition required to suit its own peculiar melting conditions. Electric steel usually needs at least twice as much addition agent as open-hearth steel to produce the same results. In spite of the exacting requirements to be met in melting treated steels, a number of mills have demonstrated their ability to produce heat after heat with at least as good uniformity as untreated steels. This has been borne out by the experience of this company and other users with a large number of heats including a number of different compositions.

Composition of Special Addition Agents

The nominal composition of a number of the currently available special addition agents is shown in Table 1. This list is similar to the one given in AISI report on "Special Alloy Addition Agents" published in November, 1942. Other compositions have been used, several of which contain vanadium. Due to the criticalness of this element, the vanadium types have been ruled out, probably for the duration.

Commercial Applications

The first commercial applications of these steels were made in the automotive industry. There are thousands of Buick motor cars on the roads today with special addition agent steels in such vital parts as steering knuckles, steering arms, knuckle supports, transmission gears, transmission main shafts, differential gears, and axle shafts. The performance of these parts has been entirely satisfactory. Other manufacturers of automotive vehicles have used addition agent steels successfully in passenger cars and trucks.

Most of the work done by this company has been on treated plain carbon and manganese steels, with carbon 0.20/0.50 and manganese in four ranges as follows: 0.70/1.00, 1.00/1.30, 1.35/1.65, and 1.60/1.90. Collectively, these steels cover an extremely wide range of properties and sections.

A recent application of more than usual interest is a long torsion bar 2 in. in diameter made of treated NE 9262 steel. This material was adopted after a number of other untreated steels were tried without success.

Table 1 – Currently Available Special Addition Agents

Alloy Designation	Al	B	Ca	Mn	Si	Ti	Zr	Fe
1	7.0	0.5	10.0	....	35.0/40.0	10.0	4.0	Balance
2	....	10.0/12.0	....	....	3.0	....	...	Balance
3	10.0/20.0	1.0/2.0	....	15.0/25.0	20.0/30.0	10.0/20.0	...	Balance
4	13.0	0.5	....	6.0	....	20.0	4.0	Balance
5	....	1.0/6.0	....	....	40.0/45.0	....	...	Balance

Fabricating Characteristics

Referring to the treated carbon and manganese steels mentioned above and comparing them with untreated steels of the same composition, the only noteworthy difference found in the shop was the improved machinability of the treated higher manganese types. In comparing the treated 1.00/1.30 manganese series with the untreated 1.60/1.90 manganese steels for which the former were substituted in a number of applications, the difference in machinability in favor of the treated steels was still more pronounced. As one member of the organization remarked, "These steels (treated 1.00/1.30 manganese) anneal and machine like carbon steels and heat-treat like alloy steels."

The forging qualities of the treated steels appeared to be as good as those of the same compositions untreated. The same was true of annealing. The GM 1340-A type, treated and untreated, responded equally well to the same annealing cycle. The treated lower manganese steels annealed more readily than the untreated higher manganese series.

Specifications

No standard specifications, such as WD, SAE, AISI, or ASTM, now exist for special addition agent steels. It has become quite obvious that chemistry alone will not suffice. The small amounts of special elements remaining from the additive treatment are very difficult to determine accurately and, moreover, their presence does not necessarily guarantee that the desired effect has been obtained. It is now apparent that hardenability and possibly mechanical properties will be required in addition to chemistry.

Tentative specifications, including chemistry, minimum hardenability, and a merit index based on tensile strength and reduction of area, have been formulated and are now receiving serious consideration.

Testing Programs

In the past few years a large amount of testing has been done on special addition agent steels by both steel producers and users. In 1942 an attempt was made to collect and correlate all of the information available. This resulted in the publication of "Contributions to the Metallurgy of Steel – No. 9," report on "Special Alloy Addition Agents" by the American Iron and Steel Institute, dated November, 1942. It is now planned to make another survey to collect whatever additional data can be obtained.

A cooperative test program involving nine basic-open-hearth heats is now in progress and a second one is being planned. These projects include a very complete program of laboratory and service tests.

Buick Laboratory Tests

All tests here reported were made in the Buick laboratories. The results given in Figs. 1 to 11 inclusive, in each case, represent treated and untreated ingots from the same heat of steel. Comparative hardenabilities for a number of treated and untreated steels are shown in Tables 2, 3, and 4.

The same addition agent was used in the treated ingots reported in Figs. 1 to 5 inclusive. This was a type containing vanadium and is not listed in Table 1. Another type of additive not containing vanadium and not appearing in Table 1 was used in the treated ingots reported in Figs. 6 and 7. Alloy No. 4 was the additive in the treated ingots of Figs. 8 and 9 and in NE 9440, Item 3, Table 3 and NE 9420, Table 4. Alloy No. 1 was used in the treated ingots of Figs. 10 and 11.

Testing Procedure

Selection of Samples – Wherever possible bars or billets from middle cuts were used. In comparing treated and untreated ingots from the same heat, samples were taken from the same location in the ingots, preferably from middle cuts. Each set of tests, including chemistry and grain size, represents a single bar or billet. No ladle analyses are reported. Sizes were 1¼ round or over.

Forging – Samples were hand-forged at 2350 F to 1¼ round for the hardenability bars and to 1 1/16 round for the tensile and Izod bars.

Annealing – After forging, samples were annealed at temperatures indicated and cooled in mica. Total time in the furnace was 1 hr.

Quenching and Drawing – Tensile and Izod bars to be drawn at 450 F were machined to 0.520 diameter and 0.470 diameter before quenching, and ground to 0.505 diameter and 0.450 diameter respectively after drawing. Tensile and Izod bars to be drawn at 900 F were quenched in the forged size of 1 1/16 diameter. After drawing, standard 0.505 diameter and 0.450 diameter bars were machined and ground for testing. Tensile results are the average of two test bars and Izod results are the average of three notches on one test bar. Total heating time in the furnace was 1 hr for quenching and 2 hr for drawing. Quenching was done in still oil at 100-120 F and still water at 70-90 F temperatures.

Hardenability bars were quenched at temperatures indicated according to standard procedure. Total heating time in the furnace was 1 hr.

Residual Elements – In a number of instances the residual elements were not recorded for either the treated or untreated samples. Where direct comparisons are made between treated and untreated ingots from the same heat, the effect of residuals is canceled out. However, it leaves some element of doubt when comparing steels from different heats. Judging from the history of the heats in question and the hardenabilities obtained on the untreated samples, it is believed that in no case did the residuals have more than a very minor influence.

P Value – A merit index, referred to as the "P value," is used to evaluate the tensile properties. It is obtained from the following formula:

$$P = \frac{T + 6R}{5}$$

where

$$P = \text{Merit index or "P value"}$$

$$T = \frac{\text{Tensile strength, psi}}{1000}$$

$$R = \text{Reduction of area, \%}$$

In deriving this formula, the tensile strength is plotted against the reduction of area and the resulting curve is assumed to be a straight line. The data upon which the original derivation was based comprised the average values for tensile strength and reduction of area of a large number of SAE alloy steels quenched in 1-in. diameter bars and drawn at temperatures of 900 to 1100 F. This assumed straight line is represented by equation (1)

$$(1) \quad T + 6R = 500 \text{ or } 500 = T + 6R$$

Equation (2) is obtained by dividing both sides of equation (1) by 5.

$$(2)\quad 100 = \frac{T + 6R}{5}$$

Equation (3) is obtained by substituting the unknown quantity, *P*, for the value of 100 on the left-hand side of equation (2).

$$(3)\quad P = \frac{T + 6R}{5}$$

A P value of 100 is, therefore, considered to be a fair average for alloy steels fully quenched in light sections and drawn at 900 to 1100 F. Higher P values denote superior and lower P values inferior tensile properties. By specifying a minimum tensile strength and a minimum P value, the relationship between strength and ductility is thus expressed by a single number.

Hardenability Index – Hardenability is expressed by means of a code or index as explained under "Method of Determining Hardenability" in the 1943 SAE Handbook, p. 323. Thus, the terms "hardenability J-50," or merely "J-50," denote the point on the hardenability curve at which the hardness is Rockwell C 50. If the distance of this point from the quenched end of the bar is, for example, 8/16 in. then the value for J-50 is 8. The reference points or J-numbers used here for different carbon contents are J-40 for 0.20 carbon, J-45 for 0.30 carbon, and J-50 for 0.40 and 0.45 carbon. These are purely arbitrary values which have been found to work satisfactorily. They are approximately 10 points Rockwell C below the maximum quenched hardness for each carbon content.

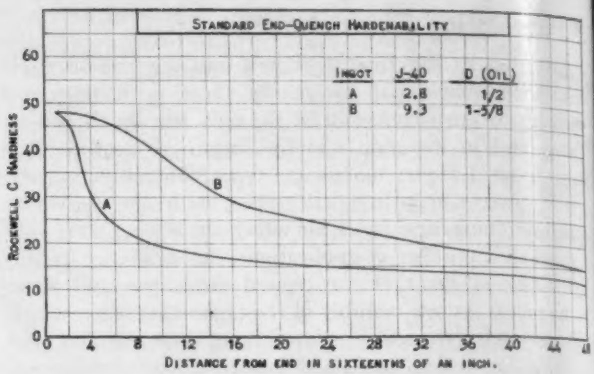
Critical Diameter (D) – The term "D," as used here, with the notation "(oil)" or "(water)" denotes the maximum diameter in inches of a round bar that will harden at the center to the hardenability index number when quenched in still oil or in still water.

■ Discussion of Results

Fig. 1 – Y-1320 – Although the specified carbon ranges are different, the samples tested conform to SAE 1024. The effect of the addition agent is very pronounced with respect to both hardenability and mechanical properties. It should be noted that the untreated sample is only partially hardened. The J-40 and the corresponding D size show that it is on the border line where it can go either way. Referring to Table 2, the treated sample exhibits a much higher J-40 hardenability than SAE 4320, SAE 4820, and NE 8720, and about the same J-40 as the treated NE 9420. Comparing Y-1320 and NE 9420, both treated and untreated, it is seen that the two steels parallel each other quite closely. The difference in carbon content must, of course, be taken into consideration.

Fig. 2 – Y-1340 – Except for the specified carbon range, this steel is a low sulphur variant of SAE 1141. In addition to a very marked improvement in hardenability and mechanical properties due to the addition agent, the treated sample has a higher J-50 hardenability than either SAE 3140 or 4140, as shown in Table 3.

Fig. 3A – GM 1340-A (450 F draw) – This series of tests was made to determine the optimum amount of additive to be used. The quantity of this particular addition agent usually recommended for basic-open-hearth steel is 4 lb per gross ton. The results indicate that this quantity



■ Fig. 1 – Test results on bars from treated and untreated ingots from a heat of Y-1320

Specification – C 0.15/0.25, Mn 1.35/1.65 – (Y-1320).
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Annealed 1650, Oil Quenched 1600 (0.520 diameter), Drawn 450.
Izod Bars – Annealed 1650, Oil quenched 1600 (0.470 diameter), Drawn 450.
Hardenability Bars – Annealed 1650, Quenched 1600.

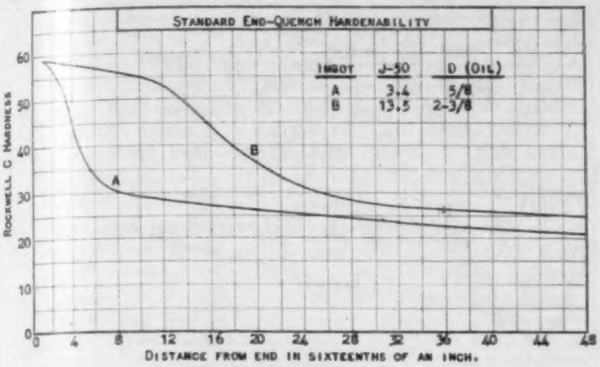
Chemical Composition and Mechanical Properties

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	7½	7½
Carbon	0.25	0.26
Manganese	1.53	1.53
Yield Point	104,300	206,300
Tensile Strength	119,900	215,600
Elongation – 2 in.	19.3	13.8
Reduction of Area	60.70	56.60
P Value	96.82	111.04
Izod Value	82.3	45.1

is the one most effective for this heat of steel. The J-50 hardenability increases progressively with increasing amounts up to 4 lb and then decreases; at 10 lb it has fallen to less than the value for 2 lb. The improvement in mechanical properties starting with the 2-lb addition is very pronounced. Referring to Table 3, the treated GM 1340-A has a slightly higher J-50 than the treated NE 9440 and a very much higher J-50 than either SAE 3140 or 4140. The 2-lb addition also has a higher J-50 than the latter two steels. The mechanical properties for the 4-, 6-, 8-, and 10-lb additions show very little difference.

Fig. 3B – GM 1340-A (900 F draw) – This is the same series as in Fig. 3A quenched in 1 1/16 diameter and drawn at 900 F. There is a progressive increase in yield point and tensile strength and a progressive decrease in Izod value with increasing amounts of additive at this draw temperature. The P values are but little different in all six samples. The untreated ingot is close to the border line in hardenability for this section and shows evidence of incomplete hardening. The lower order of P values with the 900 F draw compared with the 450 F draw is characteristic of high-hardenability steels.

Fig. 4 – GM 1340-A – These tests were made to determine the effect of small amounts of additive. This heat and the one reported in Figs. 3 (A and B) were from different sources. The J-50 hardenability shows a progressive increase with increasing amounts of additive. The im-

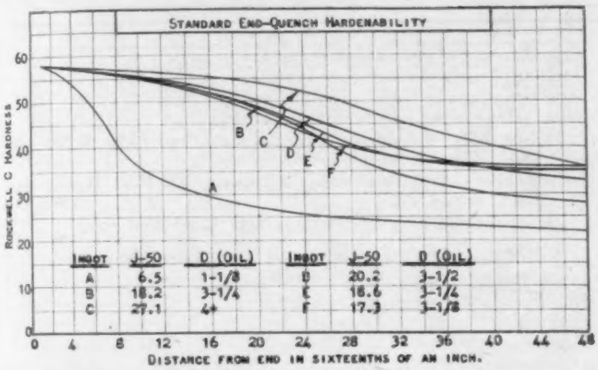


■ Fig. 2 – Test results on bars from treated and untreated ingots from a heat of Y-1340

Specification – C 0.38/0.43, Mn 1.35/1.65 – (Y-1340).
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Annealed 1600, Oil quenched 1550 (0.520 diameter), Drawn 450.
Izod Bars – Annealed 1600, Oil quenched 1550 (0.470 diameter), Drawn 450.
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	8	7½
Carbon	0.45	0.43
Manganese	1.52	1.49
Yield Point	237,000	242,000
Tensile Strength	269,600	282,800
Elongation – 2 in.	5.0	11.0
Reduction of Area	18.80	41.20
P Value	76.54	106.00
Izod Value	2.7	10.3



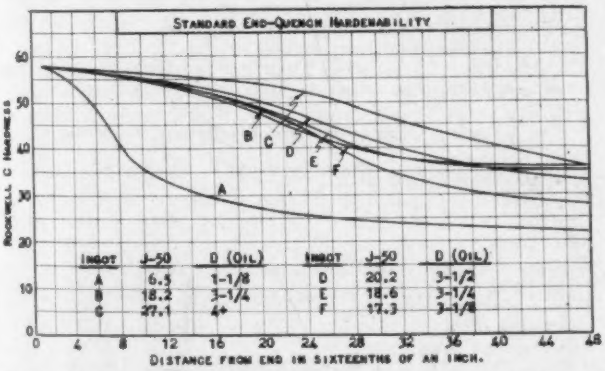
■ Fig. 3A – Test results on bars from treated and untreated ingots from a heat of GM 1340-A – drawn at 450 F

■

Specification – GM 1340-A.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Annealed 1600, Oil quenched 1550 (0.520 diameter), Drawn 450.
Izod Bars – Annealed 1600, Oil quenched 1550 (0.470 diameter), Drawn 450.
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C	D	E	F
Addition, lb per gross ton	0	2	4	6	8	10
Grain Size	7½	6	7	8	8	8
Carbon	0.40	0.42	0.41	0.41	0.41	0.41
Manganese	1.69	1.73	1.75	1.72	1.72	1.67
Yield Point	237,600	251,100	247,900	251,350	250,700	247,900
Tensile Strength	251,100	269,000	269,400	271,900	272,000	267,600
Elongation – 2 in.	9.8	12.0	13.5	12.0	13.0	12.8
Reduction of Area	33.98	41.90	51.08	47.40	48.25	49.13
P Value	91.00	104.08	115.18	111.26	112.30	112.48
Izod Value	5.5	14.8	21.8	17.8	20.8	22.0



■ Fig. 3B – Test results on bars from treated and untreated ingots from a heat of GM 1340-A – drawn at 900 F

■

Specification – GM 1340-A.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile and Izod Bars – Oil quenched 1600 (1 1/16 diameter), Drawn 900.
Hardenability Bars – Annealed 1600, Quenched 1550.

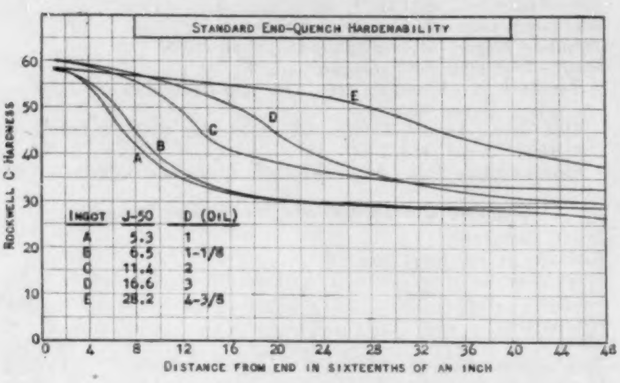
Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C	D	E	F
Addition, lb per gross ton	0	2	4	6	8	10
Grain Size	7½	6	7	8	8	8
Carbon	0.40	0.42	0.41	0.41	0.41	0.41
Manganese	1.69	1.73	1.75	1.72	1.72	1.67
Yield Point	123,600	156,300	161,500	163,100	170,700	171,800
Tensile Strength	142,900	166,600	169,400	171,900	175,900	177,300
Elongation – 2 in.	16.5	15.5	16.0	15.5	15.0	14.8
Reduction of Area	57.38	54.33	54.63	53.80	52.88	53.83
P Value	97.44	98.52	99.44	98.94	98.64	100.06
Izod Value	72.7	52.8	48.6	40.7	38.5	38.3

provement in mechanical properties follows the usual trend, but with a somewhat lower order of P and Izod values than the previous heat.

Fig. 5A – GM 5040-A (450 F draw) – This steel shows excellent response to the addition agent both in hardenability and mechanical properties.

Fig. 5B – GM 5040-A (900 F draw) – This is the same series as in Fig. 5A quenched in 1 1/16 in. diameter and

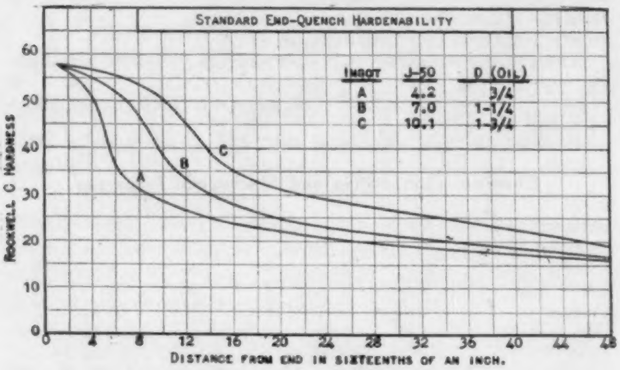


■ **Fig. 4 – Test results on bars from treated and untreated ingots from a heat of GM 1340-A**

Specification – GM 1340-A.
Melting Process – Basic open hearth.
Addition Agent – Mold addition as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Annealed 1600, Oil quenched 1550 (0.520 diameter), Drawn 450.
Izod Bars – Annealed 1600, Oil quenched 1550 (0.470 diameter), Drawn 450.
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C	D	E
Addition, lb per gross ton	0	1	2	3	4
Grain Size	7	7½	8	8	7½
Carbon	0.43	0.44	0.44	0.43	0.43
Manganese	1.74	1.76	1.71	1.71	1.71
Yield Point	247,300	248,300	247,000	244,500	242,400
Tensile Strength	270,200	273,500	273,100	274,900	279,800
Elongation – 2 in.	10.5	11.5	12.3	12.7	12.7
Reduction of Area	34.45	38.90	40.90	46.70	43.60
P Value	95.38	101.38	103.70	111.02	108.26
Izod Value	3.7	2.7	4.5	8.3	14.3

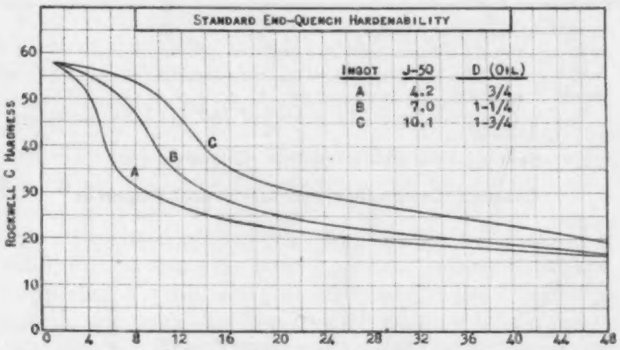


■ **Fig. 5A – Test results on bars from treated and untreated ingots from a heat of GM 5040-A – drawn at 450 F**

Specification – GM 5040-A.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Annealed 1600, Oil quenched 1550 (0.520 diameter), Drawn 450.
Izod Bars – Annealed 1600, Oil quenched 1550 (0.470 diameter), Drawn 450.
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C
Addition, lb per gross ton	0	2	4
Grain Size	7	7	7
Carbon	0.41	0.42	0.42
Manganese	0.92	0.90	0.88
Chromium	0.58	0.57	0.58
Yield Point	247,400	249,000	250,000
Tensile Strength	261,400	273,700	281,500
Elongation – 2 in.	6.7	8.5	12.5
Reduction of Area	23.95	26.30	48.55
P Value	81.02	86.30	114.56
Izod Value	7.5	11.5	14.7



■ **Fig. 5B – Test results on bars from treated and untreated ingots from a heat of GM 5040-A – drawn at 900 F**

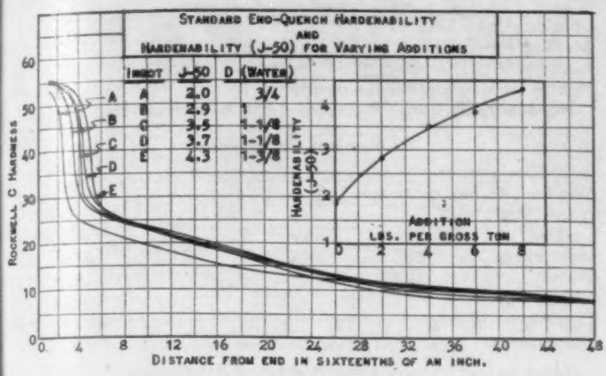
Specification – GM 5040-A.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Tensile Bars – Oil quenched 1600 (1 1/16 diameter), Drawn 900
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C
Addition, lb per gross ton	0	2	4
Grain Size	7	7	7
Carbon	0.41	0.42	0.42
Manganese	0.92	0.90	0.88
Chromium	0.58	0.57	0.58
Yield Point	106,100	142,600	168,700
Tensile Strength	132,400	155,600	176,400
Elongation – 2 in.	19.7	14.7	15.0
Reduction of Area	60.05	57.55	57.27
P Value	98.54	100.18	104.00

drawn at 900 F. The response to the additive follows the usual trend. The untreated sample is only partially hardened.

Fig. 6 – SAE 1040 – This is another series of tests made to determine the optimum amount of addition agent required. Due to the low hardenability of the base material, the tests were confined to water-quenched 1 1/16 in. diameter bars drawn at 900 F. As indicated by the J-50 values and confirmed by the tensile and Izod tests, the untreated sample and the one with the 2-lb addition are only partially hardened. The low Izod resulting from the 2-lb addition is probably associated with a critical degree of partial hardening which has been found to occur in some low-hardenability steels. Starting with the 4-lb addition, the mechanical properties are characteristic of treated steels of this type. The curve in the small box is added to show



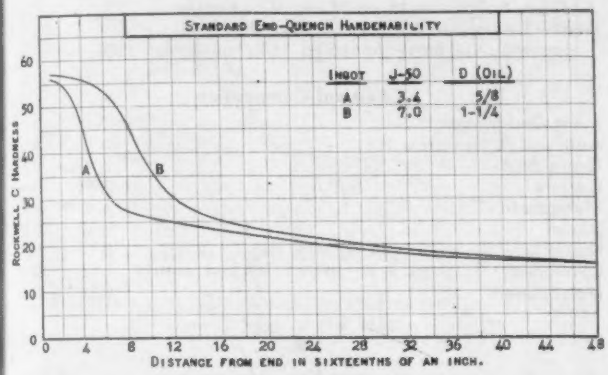
■ Fig. 6 – Test results on bars from treated and untreated ingots from a heat of SAE 1040

■

Specification – SAE 1040.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat-Treatment (All temperatures, F)
Tensile and Izod Bars – Water quenched 1600 (1 1/16 diameter), Drawn 900.
Hardenability Bars – Annealed 1600, Quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B	C	D	E
Addition, lb per gross ton	0	2	4	6	8
Grain Size	7	7	7	7½	7
Carbon	0.38	0.41	0.42	0.41	0.40
Manganese	0.75	0.75	0.75	0.75	0.76
Chromium	0.09	0.09	0.08	0.08	0.08
Yield Point	91,100	107,700	126,500	136,600	139,700
Tensile Strength	118,000	134,000	143,250	147,200	148,900
Elongation, 2 in.	22.8	20.5	16.5	17.5	18.3
Reduction of Area	57.53	62.40	55.95	57.90	60.33
P Value	92.64	101.68	95.79	98.92	102.18
Izod Value	75.8	36.8	58.5	58.3	53.8



■ Fig. 7 – Test results on bars from treated and untreated ingots from a heat containing: C 0.38/0.43 and Mn 1.00/1.30

■

Specification – C 0.38/0.43, Mn 1.00/1.30.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat-Treatment (All temperatures, F)
Tensile and Izod Bars – Oil quenched 1600 (1 1/16 diameter), Drawn 900.
Hardenability Bars – Annealed 1600, quenched 1550.

Chemical Composition and Mechanical Properties

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	7	7
Carbon	0.41	0.40
Manganese	1.15	1.17
Nickel	0.10	0.10
Chromium	0.18	0.16
Yield Point	94,900	147,100
Tensile Strength	124,300	159,500
Elongation, 2 in.	20.8	16.5
Reduction of Area	59.65	55.90
P Value	96.44	98.98
Izod Value	64.8	63.7

the J-50 hardenability plotted against the amount of addition agent.

Fig. 7 – C 0.38/0.43, Mn 1.00/1.30 – This composition does not conform to any SAE or AISI specification. With an additive treatment it is known as Buick 1341-A. It responds well to the addition agent and shows excellent mechanical properties. This steel, with addition-agent treatment, has been used in a number of high-duty automotive parts.

Fig. 8 – SAE 1045 – These results show the effect on the J-50 hardenability of a 4-lb addition to SAE 1045 and may be considered typical.

Fig. 9 – NE 8442 – Although this steel has been deleted from the NE list, it was felt that the results were of sufficient interest to justify their inclusion. A comparison with the steel in Fig. 2 gives some idea of the effect of an addition-agent treatment versus 0.28 molybdenum and the combined effect of the two.

Fig. 10 – NE 9420 – These results show the excellent response of this steel to an additive treatment. Referring to Table 2, the treated sample has a much higher J-40 than SAE 4320, SAE 4820, and NE 8720. As previously stated in discussing Fig. 1, NE 9420 and Y-1320, treated and untreated, are very similar in behavior. For the same carbon content, the NE 9420 would have a somewhat higher hardenability.

Fig 11 – NE 9430 – The response of this steel to the additive treatment is consistent with that of NE 9420.

Tables 2 and 3 are self-explanatory and have already been referred to in some detail. Table 4 shows the hardenability and core properties of untreated NE 8720 in comparison with treated NE 9420.

■ Summary

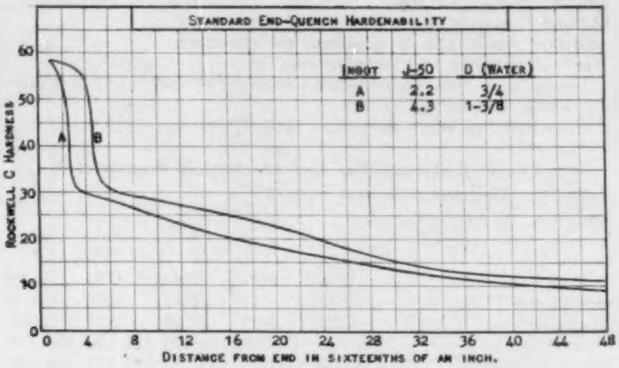
1. Certain ferroalloys containing boron, known as "special addition agents," possess the property of markedly increasing the hardenability of many steels when added in relatively small quantities.

2. These additives offer promise of conserving critical alloying elements by their ability to replace important amounts of nickel, chromium, and molybdenum.

3. The additive treatment of steel from a commercial viewpoint is relatively new, having started in 1938.

4. The general effect of additive treatment is to increase the hardenability and improve the mechanical properties of steel in the quenched and drawn state. In other than the quenched and drawn state, the mechanical properties are lowered, at least in some steels.

5. The improvement in mechanical properties from the



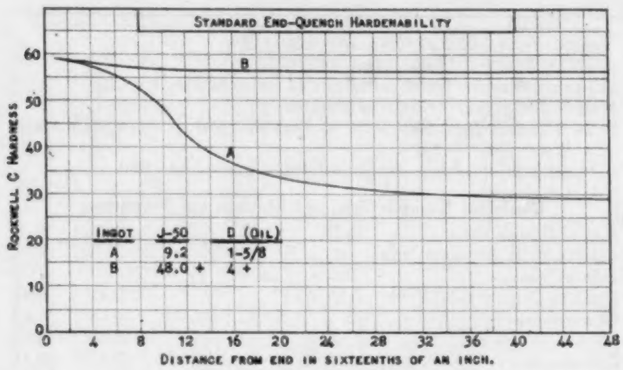
■ Fig. 8 – Test results on bars from treated and untreated ingots from a heat of SAE 1045

Specification – SAE 1045.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Hardenability Bars – Annealed 1700, quenched 1550.

Chemical Composition

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	8	8
Carbon	0.47	0.47
Manganese	0.77	0.78
Silicon	0.18	0.18
Nickel	0.05	0.06
Chromium	0.03	0.04
Molybdenum	Nil	Nil

■



■ Fig. 9 – Test results on bars from treated and untreated ingots from a heat of NE 8442

Specification – NE 8442.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Hardenability Bars – Annealed 1600, quenched 1550.

Chemical Composition

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	8	8
Carbon	0.45	0.45
Manganese	1.45	1.45
Silicon	0.32	0.33
Nickel	0.12	0.16
Chromium	0.06	0.07
Molybdenum	0.28	0.27

■

Table 2 – Hardenability of 0.20 Carbon Steels Treated and Untreated

Item No.	Specification No.	T or U*	Number of Heats	C	Mn	Si	Ni	Cr	Mo	Grain Size	Hardenability (J-40)	D (Oil)
1	SAE 4320	U	8	0.20	0.55	....	1.82	0.52	0.23	7.4	4.1	¾
2	SAE 4820	U	10	0.21	0.54	....	3.57	....	0.22	6.8	5.5	1
3	NE 8720	U	8	0.22	0.81	0.22	0.56	0.49	0.23	8.0	3.4	⅝
4	NE 9420	U	1	0.21	0.99	0.48	0.48	0.35	0.12	8.0	2.7	½
5	NE 9420	T	1	0.20	0.98	0.57	0.47	0.35	0.12	8.0	9.6	1¾
6	Y-1320	U	1	0.25	1.53	....	....	....	....	7.5	2.8	½
7	Y-1320	T	1	0.26	1.53	....	....	....	....	7.5	9.3	1⅝

* T or U indicates treated or untreated.
Values for Items 1, 2, and 3 are averages of 8, 10, and 8 heats respectively.
Items 4 and 5 are from Fig. 10 – treated in ingot mold.
Items 6 and 7 are from Fig. 1 – treated in ingot mold.
All heats are basic open hearth.

additive treatment is confined mostly to the carbon and low-alloy steels, and is most pronounced at draw temperatures of 300-500 F.

6. In general, it may be stated that, with respect to hardenability and mechanical properties, a carbon steel can be made equivalent to a low-alloy steel and a low-alloy steel equivalent to a high-alloy steel by additive treatment.

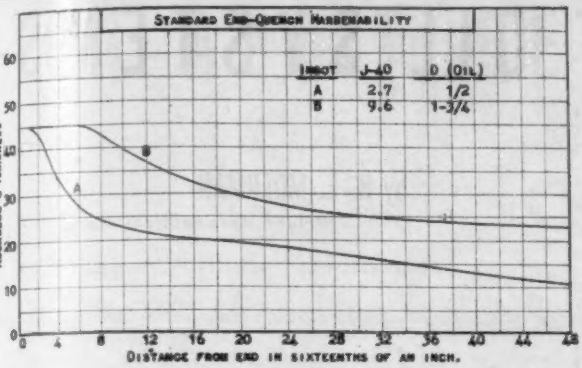
7. The addition agents may be added either in the ladle or ingot mold, preferably in the ladle. Ingot mold additions greatly facilitate experimental work.

8. The amount of additive required varies, depending upon the type of additive, the composition of the steel and the degree of deoxidation. Uniform melting practice is essential to good results.

9. If increasing additions of a given type of additive are made to different ingot molds of the same heat, the hardenability will be found to increase with the amount of additive up to a maximum or optimum value, and then decrease with further increases of additive. Maximum mechanical properties usually coincide with maximum hardenability. Additions greater than the optimum tend to lower the mechanical properties.

10. In spite of the exacting requirements to be met in melting treated steels, a number of mills have demonstrated their ability to melt heat after heat with as good uniformity as untreated steels. This has been borne out by the experience of this company and other users with a large number of heats including a number of different compositions.

11. The work done by this company has been mostly on 0.20/0.50 carbon steels with four manganese ranges as follows: 0.70/1.00, 1.00/1.30, 1.35/1.65, and 1.60/1.90. Collectively, these steels cover an extremely wide range of properties and sections.



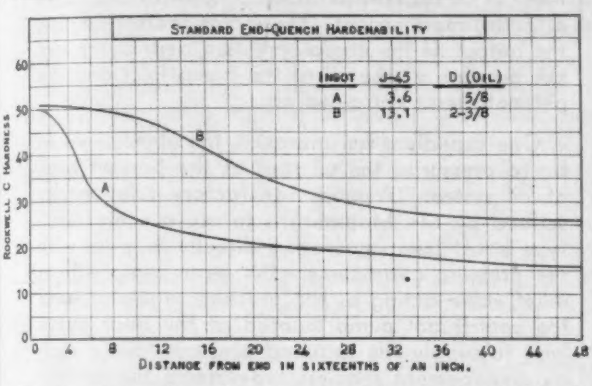
▪ Fig. 10 – Test results on bars from treated and untreated ingots from a heat of NE 9420

Specification – NE 9420.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatemnt (All temperatures, F)
Hardenability Bars – Annealed 1700, quenched 1700.

Chemical Composition

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	8	8
Carbon	0.21	0.20
Manganese	0.99	0.98
Silicon	0.48	0.57
Nickel	0.48	0.47
Chromium	0.35	0.35
Molybdenum	0.12	0.12

■



▪ Fig. 11 – Test results on bars from treated and untreated ingots from a heat of NE 9430

Specification – NE 9430.
Melting Process – Basic open hearth.
Addition Agent – Mold additions as indicated.
Heat – Treatment (All temperatures, F)
Hardenability Bars – Annealed 1700, quenched 1600.

Chemical Composition

Ingot Designation	A	B
Addition, lb per gross ton	0	4
Grain Size	8	8
Carbon	0.33	0.33
Manganese	1.02	1.03
Silicon	0.48	0.57
Nickel	0.50	0.49
Chromium	0.26	0.25
Molybdenum	0.12	0.12

■

Table 3 – Hardenability of 0.40 Carbon Steels Treated and Untreated

Item No.	Specification No.	T or U*	Number of Heats	C	Mn	S	Ni	Cr	Mo	Grain Size	Hardenability (J-50)	D (Oil)
1	SAE 3140	U	10	0.41	0.82	....	1.20	0.66	....	8.0	8.0	1⅜
2	SAE 4140	U	10	0.40	0.80	....	....	0.97	0.18	7.2	11.9	2⅛
3	NE 9440	T	1	0.40	1.06	0.53	0.44	0.30	0.12	8.0	24.0	4
4	GM 1340-A	U	1	0.40	1.69	....	....	....	....	7.5	6.5	1⅛
5	GM 1340-A	T	1	0.41	1.75	....	....	....	....	7.0	27.1	4+
6	Y-1340	U	1	0.45	1.52	....	....	....	....	8.0	3.4	⅝
7	Y-1340	T	1	0.43	1.49	....	....	....	....	7.5	13.5	2⅜

* T or U indicates treated or untreated.
Values for items 1 and 2 are averages of 10 heats each.
Item 3 was treated in the ingot mold.
Items 4 and 5 are from Figs. 3A and 3B (Ingots A and C) – treated in ingot mold.
Items 6 and 7 are from Fig. 2 – treated in ingot mold.
All heats are basic open hearth.

12. The treated steels referred to in paragraph 11 fabricate at least as well as the same steels untreated.

13. No standard specifications now exist for special addition agent steels. It is obvious that chemistry alone will not suffice, and that hardenability and possibly mechanical properties must be specified in addition to chemistry. Tentative specifications including these three requirements have been formulated and are now receiving serious consideration.

14. Test data pertaining to treated steels have been collected and published in an AISI report. Another survey for additional data is being planned.

15. A cooperative test program involving nine heats of treated steel is now in progress and a second program is about to be inaugurated.

16. Buick laboratory tests described herein show the effect of varying amounts of additives and comparative properties of a number of treated and untreated steels.

Table 4 – Pseudocarburizing Tests on NE 8720 Untreated and NE 9420 Treated

Specification	NE 8720	NE 9420
Number of Heats	8	1
Grain Size	8	7
Hardenability (J-40)	3.4	6.4
D (Oil)	⅝	1⅛
Carbon	0.22	0.20
Manganese	0.81	0.89
Silicon	0.22	0.22*
Nickel	0.56	0.29
Chromium	0.49	0.33
Molybdenum	0.23	0.13
Yield Point	165,700	167,500
Tensile Strength	194,500	188,300
Elongation – 2 in.	12.5	14.8
Reduction of Area	47.83	61.3
P Value	96.30	111.22

* This is the modified type of 9420 with silicon 0.20/0.35.
All heats are basic open hearth.
Values for NE 8720 are averages of 8 heats.
NE 9420 treated with ladle addition.
Heat-treatment (All temperatures, F).
Tensile Bars – Annealed 1700, Pseudocarburized in pitch coke at 1700 for 8½ hr and oil quenched from box (0.520 diameter), drawn 300.
Hardenability Bars – Annealed 1650, Quenched 1600.

AIRCRAFT OIL SYSTEMS—HI

by H. E. MOERMAN
Wright Aeronautical Corp.

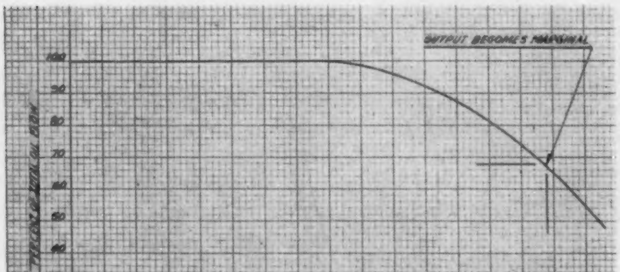
THE modern high-output aircraft engine and the requirement to fly it at altitudes beyond 30,000 ft have introduced problems in the aircraft oil system that heretofore did not require serious consideration. As a result of recent research on various types of airplane oil systems and also work done on various engine oil pumps, it has been found that the pumping characteristic of the engine pressure gear pump is largely dependent on the condition of the oil at the pump inlet. It is apparent that the major factors limiting the output of the gear pump are the pressure of the oil at the pump inlet and the percentage of entrained air. It is intended herewith to review these factors from the standpoint of their effect on pump performance at high altitudes.

■ Oil Inlet Suction

Fig. 1 shows the variation in output obtained on an aircraft oil pump when the oil has 5% of entrained air. It is plotted against altitude, or atmospheric pressure at the pump inlet. The curve indicates a reduction in pump output when the inlet pressure becomes less than the atmospheric pressure equivalent to 20,000 ft. This pump is used on an engine requiring a given oil flow such that the pump capacity does not become marginal until its inlet is at 37,000 ft equivalent altitude. For altitudes above this point, pump output becomes less than the desired value and the oil pressure will begin to fall off. At some higher altitude, the pressure will reach a value low enough to result in insufficient lubrication.

Many oil systems have been designed which produce an oil inlet suction as high as 6 in. Hg. If this suction value is permitted, the absolute pressure at the pump inlet will be equivalent to 37,000 ft when the airplane is flying at 22,500 ft. Thus the oil system line loss reduces the oil pump critical altitude approximately 15,000 ft. Obviously, 6 in.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 8, 1943.]



■ Fig. 1 – Variation of oil flow with altitude for a typical aircraft gear pump with 5% entrained air

THE use of high-output aircraft engines at altitudes above 30,000 ft has made the development of oil systems suitable for operation at these altitudes most urgent. The major factors limiting the output of the engine pressure gear pump are the pressure of the oil at the pump inlet and the percentage of entrained air.

One expedient for improving the altitude would be to pressurize the oil tank of the present type of oil system. A more satisfactory solution described by Mr. Moerman is to use a centrifugal-type boost pump operating directly in series with the pressure gear pump – the gear pump with a relief valve acting as the pressure regulator, and the centrifugal pump located at the gear pump inlet to supply the required pressure. Since such an arrangement requires redesigning the present oil pump systems, an alternate arrangement whereby the centrifugal pump is mounted at the oil tank could be employed.

Air entrainment must also be studied in relation to oil pump output, for the presence of an excessive amount of air in the oil is manifested by loss of oil pressure, oil pressure fluctuation, and foaming in the oil tank. Recent testing indicates that the centrifugal type of separator is very effective for separating entrained air. This separator, Mr. Moerman explains, is essentially a more precise application of the tangential-entry principle than has heretofore been used.

■ ■ ■

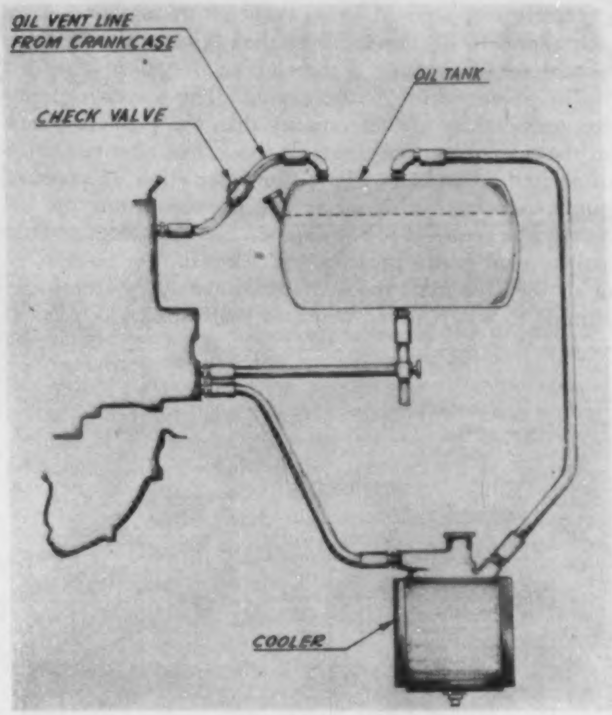
THE AUTHOR: H. E. MOERMAN, as project field engineer for Wright Aeronautical, establishes methods of installation design, test and operation for all Wright Cyclone 9 and Cyclone 14 engines. Before his present assignment he worked on aircraft-engine installation design and test in the Field Engineering Division. His work with Wright began in the experimental test laboratory as observer and engine tester, after which he became carburetor test engineer.

Hg inlet suction cannot be tolerated in oil systems which are intended for high-altitude aircraft. The suction should be kept to an absolute minimum in order that the oil system itself imposes little or no penalty in pump output. It is, therefore, necessary to provide an oil inlet line of ample size with a minimum of turns, elbows, and miscellaneous fittings. In high-altitude oil system design it appears desirable to limit the inlet line loss to 20% of the atmospheric pressure at the altitude involved.

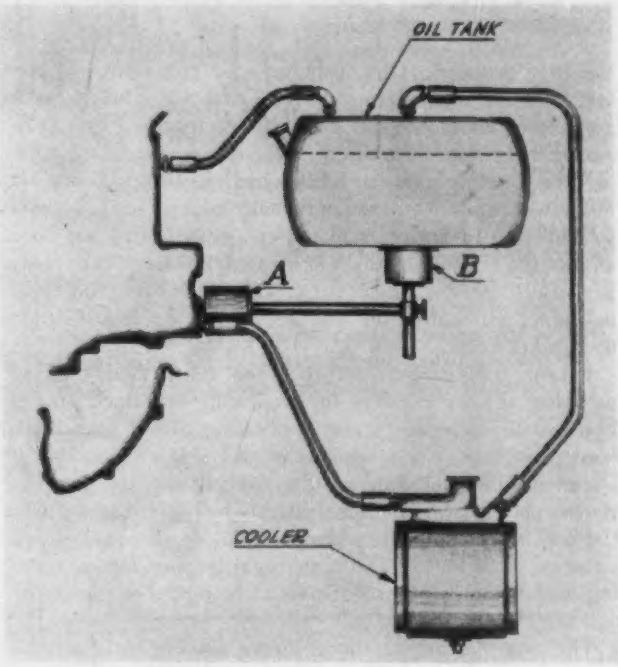
■ Increasing Altitude Performance

A simple expedient for improving the altitude performance in the present type of oil system would be to pressurize the oil tank. Such an arrangement is shown in Fig. 2. This system provides a check valve in the oil tank vent line between the tank and the engine. With this system the engine scavenge pump serves as the pressure supply for the oil tank. Scavenge oil from the engine contains a large amount of air, and since the engine scavenge pump is approximately twice the capacity of the pressure pump, it is capable of providing pressure boost for the oil tank. To demonstrate the possibilities of tank pressurization, let us assume that the pump which becomes marginal at 37,000 ft, or 6.4 in. Hg inlet pressure, is installed in a theoretical oil system which has an inlet line loss of 20% of atmospheric pressure. The pump inlet pressure would be 6.4 in. Hg when the atmospheric pressure is 7.7 in. Hg or 33,000 ft. If the check valve is set to maintain 2 psi pressure in the oil tank, this altitude would be raised to approximately 50,000 ft. Although pressurizing the oil tank appears to be an easy expedient for increasing the oil system critical altitude, there appear to be some objections to the use of such a system on a combat-type airplane where bulletproof cells are employed in the tanks. This item is subject to some questions, however, and it is entirely possible to use armor for oil tank protection rather than employing self-sealing cells within the tank.

A more satisfactory solution to the entire problem appears possible by the use of a centrifugal-type boost pump operating directly in series with the pressure gear pump. This centrifugal pump would be driven from the same shaft as the gear pump and would supply oil to the gear pump under pressure. It is well known that the centrifugal type of pump, by virtue of its inlet design, is capable of operating satisfactorily with almost zero absolute inlet pressure. Therefore the centrifugal pump is a much more desirable pump for receiving oil from the airplane oil system than is the gear-type pump. On the other hand, however, it is not possible to discard the gear pump in favor of the centrifugal pump since the engine requires closely controlled oil pressure regulation, and the gear type of pump is more adaptable to such regulation. This basic problem of pump characteristics has previously been encountered in non-aircraft pump installations. The usual remedy has been to use a gear pump with a relief valve as a pressure regulator, and a centrifugal pump as a booster to supply the required pressure at the gear pump inlet. This application could be made to aircraft installations as shown on Fig. 3. The booster pump could be made of sufficient capacity to provide approximately 15 in. Hg boost at rated speed. Since such an arrangement requires an entire redesign of the present engine oil pump system, an alternate arrangement whereby the centrifugal boost pump is mounted at the oil tank would have to be employed for



■ Fig. 2 – Pressurized tank



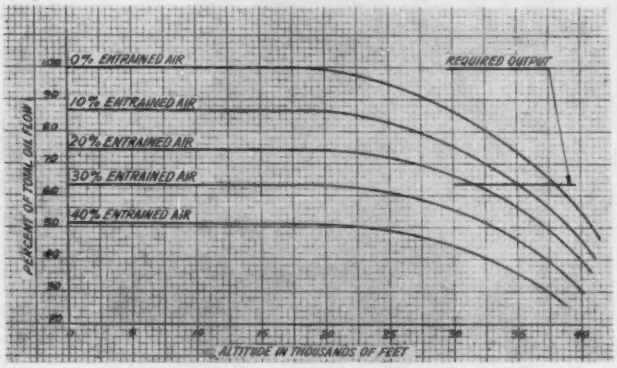
■ Fig. 3 – Centrifugal pump installations: *A* – centrifugal booster integral with the engine pump; *B* – Centrifugal booster at oil tank outlet

high-altitude systems using present-type engines. This alternate system is also shown on Fig. 3. In this location the pump would have to be electrically driven. However,

it appears that the electrical requirements for such a pump would become excessive when attempting to pump oil at cold temperatures.

■ Air Entrainment

Closely related to oil pump output is the problem of air entrainment. Oil removed from the engine by the scavenge pump contains a large percentage of air which is picked up in passing through the engine. The scavenge pump has considerably greater capacity than the pressure pump so the oil returned to the tank consists of alternate slugs of aerated oil and air. Fig. 4 shows the effect on pressure pump capacity if this air is not removed from the oil before it is returned to the engine. For zero entrained air a theoretical pump capacity was selected. For each 10% of air the capacity is reduced approximately 13%, and the altitude to which it can maintain that capacity is propor-



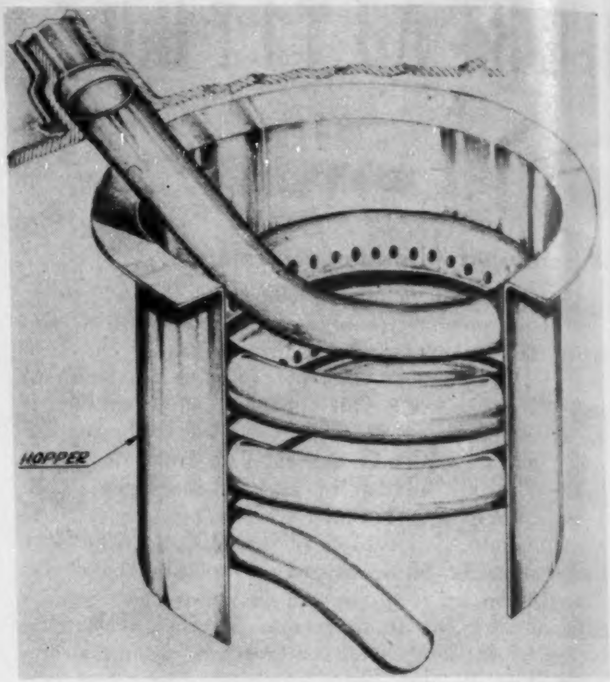
■ Fig. 4 – Effect of entrained air on oil pump capacity

tionately reduced. Obviously it is of paramount importance that the air be removed. The presence of an excessive amount of air in the oil is manifested by loss of oil pressure, oil pressure fluctuation, and foaming in the oil tank.

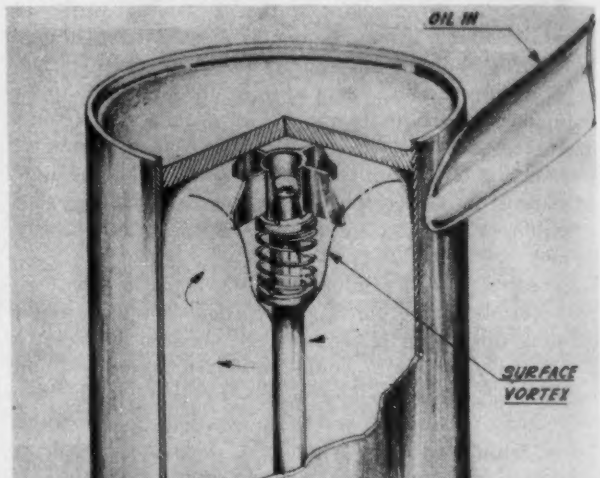
Various expedients have been used in the past and are fairly well known. Most of them consist of some type of splashplate. In hopper tanks, a tangential entry has been almost universally used. These methods have been only partially successful, and a better solution to the problem is desirable, especially due to the advent of high-altitude operations.

Recent testing indicates that the centrifugal type of separator is very effective for separating entrained air. It is essentially a more precise application of the tangential-entry principle. Early versions of this separator consist of a coiled-tube extension of the tangential-entry pipe as shown on Fig. 5. The tube extends helically down inside the top of the hopper. Along its inside diameter are a series of small holes. The centrifugal force of the oil turning within the coil causes the air to be forced to the inside and out through the holes.

The Sharples air separator shown on Fig. 6 is a commercially available device which assists in removing entrained air from the oil. It is mounted in the oil outlet line between the scavenge pump and the oil cooler. It consists of a small cylindrical tank with tangential oil inlet and outlet connections and a vent for separated air. The small tube along the centerline of the cylinder is the air vent connection. The top of this tube is closed except for a small hole drilled off center. A small four-bladed rotor



■ Fig. 5 – Centrifugal air separator – coiled-tube type



■ Fig. 6 – Sharples air separator

fits over the top of the tube and has a hole that matches the one in the tube. A light spring holds the rotor against

turn to page 407

NOTES on the SELECTION and Installation of AIRCRAFT PROPELLERS

by THOMAS B. RHINES

Hamilton Standard Propellers Division, United Aircraft Corp.

AIRCRAFT propeller selection and installation is, by nature, a problem of coordination and cooperation among several separate organizations. The propeller manufacturer, airplane manufacturer, and engine manufacturer are always involved, and in many cases the vendors of accessory equipment may influence the ultimate propeller selection for an airplane. In this paper, some of the problems that require the cooperation of the airplane and propeller manufacturers in particular are considered.

A brief outline of some of the points that must be evaluated in propeller selection is useful to clarify this point. While the considerations involved are many, the following are among the most important:

1. Aerodynamic performance.
2. Propeller weight and size.
3. Airplane-engine-propeller vibration characteristics.
4. Detailed installation requirements.
5. Propeller structural loads.
6. Operating loads.
7. Blade-angle-range requirements.
8. Rate of angle change.
9. Availability in production.

Several of the above categories are primarily the concern of the propeller manufacturer. A particular example lies in the consideration of structural loads; full responsibility for the strength of the propeller lies with him. An appreciation of the problems involved is, however, helpful in an airplane organization to avoid extensive investigations of propellers that might be aerodynamically suitable, but are structurally not acceptable for the required installation. In most cases, it is impractical to assign ratings for propeller speed and power to be used as a final guide for the airplane designer, and the direct assistance of the propeller manufacturer therefore becomes necessary.

A consideration of the forces required to operate a variable-pitch propeller presents a similar situation, but with the added complication that the operating loads are intimately associated with the rate at which the propeller blade angles can be made to change. The rate of angle change itself will be of interest to the airplane man, because it is a measure of the ability of the propeller to provide satisfactory constant-speeding characteristics under such adverse conditions as are often encountered in military aircraft.

[This paper was presented before a meeting of the Wichita Section of the SAE, Wichita, Kan., May 5, 1943.]

AIRCRAFT propeller selection and installation by the airplane designer are best accomplished with the full cooperation of the propeller manufacturer and the engine manufacturer. This period of cooperation should start immediately after the airplane general arrangement has been chosen, approximations to the principal performance characteristics have been made, an engine or a small group of possible engines has been selected, and an indication is available of the general space that can practically be made available for the propeller.

The designer, however, should not wait until this stage to consider the general propeller requirements of the airplane. He should make some preliminary calculations for himself, so that he can be sure that there are propellers that will fill his requirements.

When the propeller manufacturer is consulted, he will normally make his recommendations as a group of possible propellers, any one of which would probably be suitable for the airplane as far as structural strength, angle range, and availability are concerned, but which show variations at the different operating conditions, and differences in weight, diameter, and other properties.

These recommendations will be accompanied by performance estimates for the principal operating conditions that will have to be extended by the airplane designer to cover a wider variety of flight conditions. Methods for making such propeller performance estimates are discussed by Mr. Rhines.

From these estimates, the best propeller is selected and the installation details are definitely settled, again in consultation with the propeller manufacturer, who should review the final selection thoroughly.

THE AUTHOR: THOMAS B. RHINES (M '38) has been author or co-author of several papers presented before the SAE on various subjects, including the cooling of aircraft engines and phases of aircraft propeller performance. Mr. Rhines since 1939 has been with Hamilton Standard Propellers Division, and is now chief production engineer. Prior experience was with the Research Division of United Aircraft, where he went as aeronautical engineer after graduation from Massachusetts Institute of Technology.

As illustrative of this point, it might be noted that for a typical 15-ft three-blade propeller turning at 1350 rpm, the angle change required between a closed-throttle windmilling condition and a condition of full power is on the order of 10 deg. To cope fully with a throttle burst from closed throttle to maximum opening in 1 sec, with no appreciable overspeed, the rate of pitch change should approach 10 deg per sec, and the power available from the propeller operating mechanism to achieve this should be on the order of 3 hp. Present production propellers in the United States are available with usable pitch-change rates from approximately 1 to 10 deg per sec; in a choice among them the probable required pitch-change rate should not be overlooked.

In the list I have given, there are other items that are of more direct and active interest to the airplane designer. The most obvious of these are the propeller performance, its weight and size, and its installation requirements. In general, the consideration of at least these items should be the subject of intensive cooperation between propeller and airplane organizations. The mechanism for providing effective cooperation is apparent, but its use to the best advantage is a matter involving a large measure of judgment. A propeller manufacturer cannot be of much assistance before an airplane design has begun to take shape to the point where the general design objectives have been clarified. Neither can the propeller man be helpful when the design has been completely frozen and a propeller chosen. It is necessary to seek a particular point in the creation of the airplane where cooperation will be most effective.

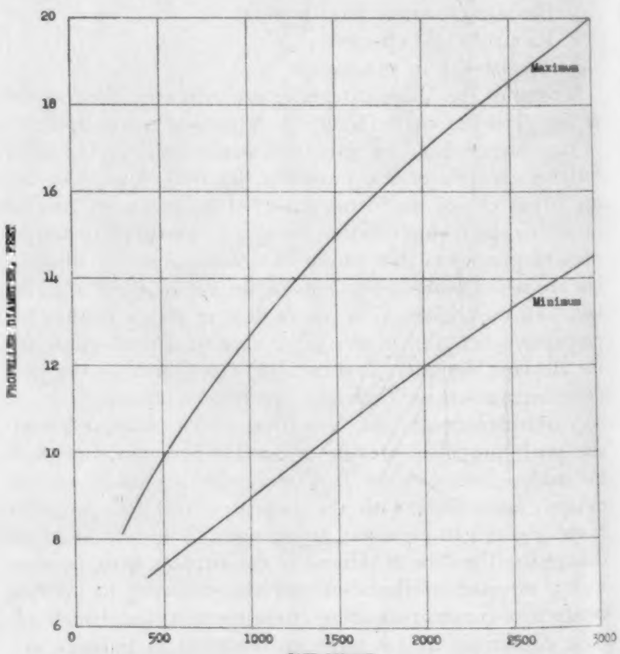
As a guide to the selection of this point, it can best be described as the stage in which the airplane general arrangement has been chosen, approximations to the principal performance characteristics have been made, an engine or a small group of possible engines has been selected, and an indication is available of the general space that can practically be made available for the propeller. From these characteristics, it should be possible to provide the propeller designer with a list of the important airplane flight conditions, including speeds and the related altitudes and engine powers, and a general discussion of the relative importance of the various performance requirements. The latter is a particularly valuable contribution to effective cooperation. In general, a propeller that is best for take-off will differ appreciably from a propeller that is best for maximum speed, and there are similar conflicts that must be resolved among all of the principal performance requirements. In seeking a best compromise, numbers alone are not sufficient. To provide a propeller for best top speed on an airplane that is suffering from a marginal single-engine climb performance is, for example, the possible result of an inadequate discussion with the propeller designer.

Before an airplane design has developed to a point where cooperation of this nature can become effective, the airplane designer should, within his own organization, explore the general propeller requirements. To overlook the propeller even at the initiation of a design problem, can too easily lead to ineffective design studies which must later be scrapped because there are available no propellers that can possibly fill the resulting requirements. The principal symptoms of this disease is usually the provision of too little clearance for an adequate propeller diameter. There can, however, be less obvious symptoms, such as failure to remember that propeller blade width may affect cowl location and cowl shape to a degree sufficient to affect airplane drag.

Propeller studies in this stage must necessarily be general, because even the power of the engine to be used may be unsettled. Charts and tables for generalized propeller information can be made up in a variety of forms that are useful for this class of work, and probably each airplane designer will suit his own fancy. Fig. 1 is about the simplest type that could be used for such preliminary studies. It is purely a record of experience based on a wide variety of actual installations, and its limitations are obvious. Another form of generalization appears in Figs. 2 and 3. In this case, propeller diameter can be blindly chosen as a function of generalized propeller characteristics and airplane operating conditions. Figs. 4 and 5 are typical general studies for specific operating conditions, but for hypothetical propellers. They show at a glance the importance of propeller size in determining propeller performance. They assume, however, that an ideal propeller reduction gear is to be available with the engine, and this is never the case. Such curves, in fact, assume that a different gear is available for each operating condition of the airplane.

These figures are illustrative of the type of thing that can be done and is usually done for preliminary evaluations. All such methods fail when specific propeller selections are attempted, simply because the variety of propellers available is insufficient to match every possible point on the curves, and because certain propellers that do appear to match may be unsatisfactory as regards availability, angle range, or in some other important aspect.

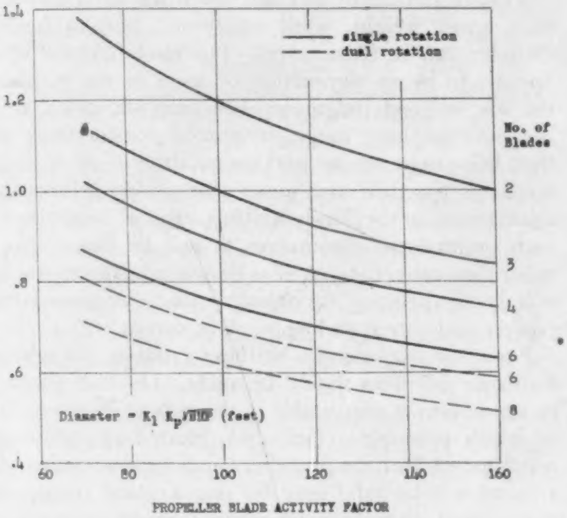
This will, then, represent the point at which one or more propeller manufacturers should be consulted for preliminary propeller recommendations to be based on as complete information as can be made available to him. His selections will normally consist of a group of possible pro-



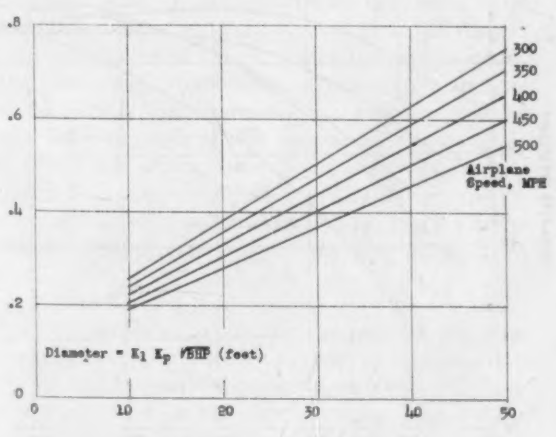
■ Fig. 1 – Estimated range of diameters for single rotation propellers with various take-off horsepowers

pellers, any of which might be suitable for the airplane, but which show variations in suitability at the different operating conditions, and differences in weight, diameter, and so forth. All of the propellers will, however, be classed as suitable as regards structural strength, angle range, and availability, or at least the recommendations will be accompanied by a discussion of these points. In most cases, suitability from the point of view of the vibration characteristics of the proposed combination will be an unknown, and must later be examined in ground tests of the engine-propeller combination and by flight tests in the prototype airplane. A preliminary evaluation will have been made to eliminate from consideration any propellers that are known to be unsatisfactory as regards vibration on the specified engine.

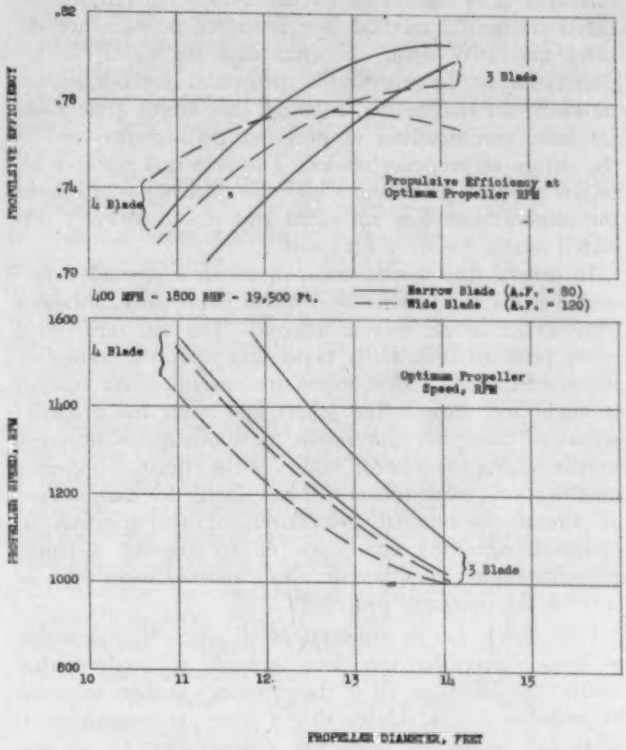
The propeller recommendations will be accompanied by performance estimates for the principal airplane operating conditions, or in general for whatever conditions estimates have been requested. These estimates will not be sufficient for a complete aerodynamic analysis of the airplane and the manufacturer will want to extend them to a wide variety of flight conditions. The methods of propeller



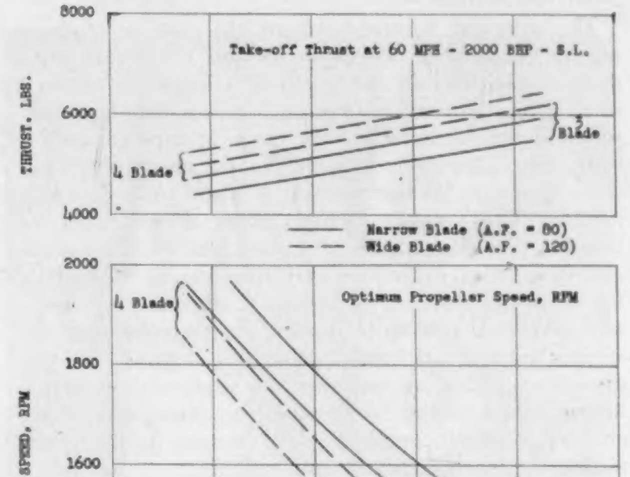
■ Fig. 2 – K_p factor for estimating propeller diameter requirements



■ Fig. 3 – K_1 factor for estimating propeller diameter requirements



■ Fig. 4 – Estimated performance of three- and four-blade propellers to give 400 mph at 19,500 ft with 1800 bhp



■ Fig. 5 – Estimated performance of three- and four-blade propellers to give 60 mph take-off at 2000 bhp (sea level)

performance estimation to be used for this purpose merit some discussion.

For practical purposes, all of the methods in current use are empirical in nature and in general all depend for the principal part of their basic data upon the propeller per-

formance tests conducted by the NACA. Even the so-called theoretical methods for propeller performance depend ultimately upon empirical data for airfoil section characteristics. In general, the theoretical methods are too unwieldy for routine use, although they are of great value for basic investigations of propeller performance and in the design of propeller blades. The principal methods all follow a typical pattern in which the efficiency is estimated for conditions of low tip speed and is subsequently corrected to the operating tip speed.

In general, the specific form in which a method is presented is not fundamentally of great importance, although convenience of use may be affected. The best form for a given problem is basically dependent upon the nature of the problem itself. Some forms are convenient for analysis of flight test data, while others are better for propeller selection. Some are convenient for comparisons among a variety of propellers or for studies of the effects of propeller modifications, while others will be valuable for examination of aircraft operational procedures, or the selection of optimum operating conditions of an existing airplane-propeller-engine combination. Still another form may be best for the design of propellers.

For general use in problems of all types, it is desirable to reduce propeller test data through an analysis that isolates the effects of all of the primary variables involved in propeller design. Unless this is done, erroneous conclusions can be drawn from test results obtained on one specific propeller and applied to another, except, of course, in those rare cases where a test exists on the actual propeller that is to be used.

The principal recommendations that can be made as regards choice of a calculation method are first to avoid those that depend on the results of a single test or on a small group of tests as opposed to those that have consolidated the results of a wide range of experimental investigations. Secondly, it is well to use one method consistently as long as that method is found to be generally reliable. This second recommendation extends from the fact that propulsive efficiency is not a true efficiency at all, and its absolute magnitude may therefore be misleading. For successful handling of propulsive efficiency, it is necessary to correlate it with airplane drag estimates in the accumulation of data from airplane drag analyses. The two elements together are necessary for airplane performance determination, and if the methods for estimating one are changed without complementary changes in the other, misleading conclusions will result.

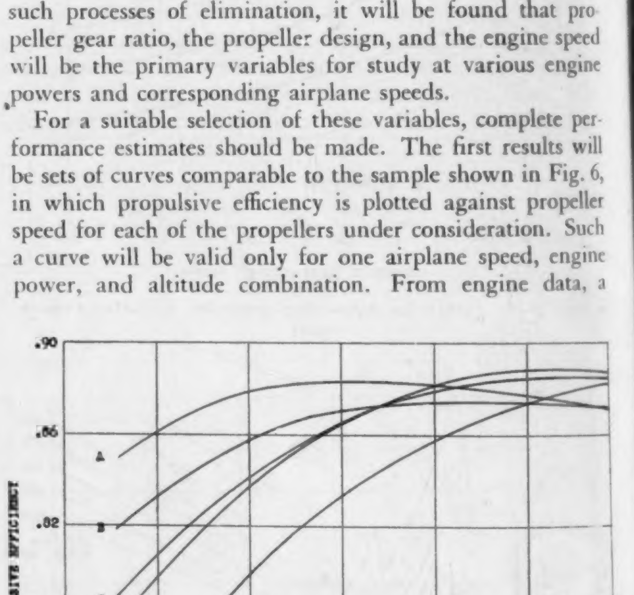
The present state of development of calculation methods is directly limited by the character of available test data. The most useful tests have been run at low tip speeds, while almost all actual propeller requirements involve high tip speeds. The methods currently available for correction from one condition to the other are not too satisfactory. Use of low-speed data without any correction may, however, be very misleading, and an attempt at correction must therefore be made. As more complete tests are accomplished, this situation will be improved; meanwhile each bit of practical experience should be used to the utmost as background for the problem. The propeller manufacturers can serve a useful purpose in this connection by recommending against propeller selections that appear to them to conflict with their accumulated experience.

Regardless of the method of propeller performance calculation that is chosen, the pattern for a propeller analysis is not much affected. Solutions for certain operating conditions are relatively straightforward, as exemplified by efficiency at top speed and propeller thrust for take-off. These are straightforward simply because the operating conditions are relatively invariable: A given engine will normally have only a single take-off rating, and top-speed determinations will be necessary for only a relatively few engine operating conditions over a range of altitudes. For analyses in the cruising region, the requirements are considerably more complex, and for a thorough analysis of even a single engine-airplane-propeller combination, aerodynamics organizations should plan a good many hours of work.

Numerous attempts have been made to provide short-cut and approximate methods for this type of study, but unfortunately even the most complete analyses are somewhat too approximate, and in general no satisfactory short cuts have yet been developed. The long, hard way can briefly be described as taking all possible operating conditions and carrying through complete calculations for each, comparing all of the resulting performance points to find the best conditions for each propeller, and finally making a comparison among all of the propellers at those best conditions.

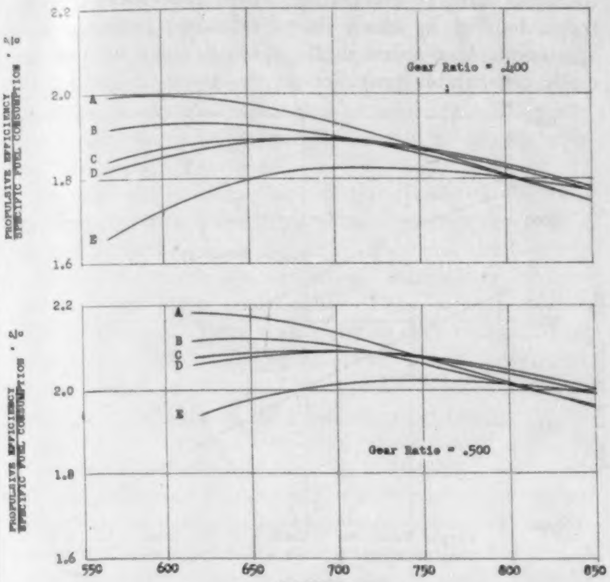
The magnitude of this full job when variations in altitude, gross weight, wind conditions, and so forth, are included can be tremendous. The most practical solution appears to be an elimination of some of the variables by the use of good judgment. For example, in an airplane designed for long range, if general consideration shows that the range at the maximum altitude to be used in service is less than at a lower altitude, propeller selection calculations at the lower altitude can be minimized. By such processes of elimination, it will be found that propeller gear ratio, the propeller design, and the engine speed will be the primary variables for study at various engine powers and corresponding airplane speeds.

For a suitable selection of these variables, complete performance estimates should be made. The first results will be sets of curves comparable to the sample shown in Fig. 6, in which propulsive efficiency is plotted against propeller speed for each of the propellers under consideration. Such a curve will be valid only for one airplane speed, engine power, and altitude combination. From engine data, a



■ Fig. 6 – Comparative performance of several propellers in cruising at 220 mph, 1000 bhp, and 25,000 ft

corresponding curve of specific fuel consumption against propeller speed must be available for each gear ratio that is under consideration. The efficiencies and fuel consumptions are then combined as indicated in Fig. 7.



■ Fig. 7 – Comparative range parameters of several propellers in cruising at 220 mph, 1000 bhp, and 25,000 ft

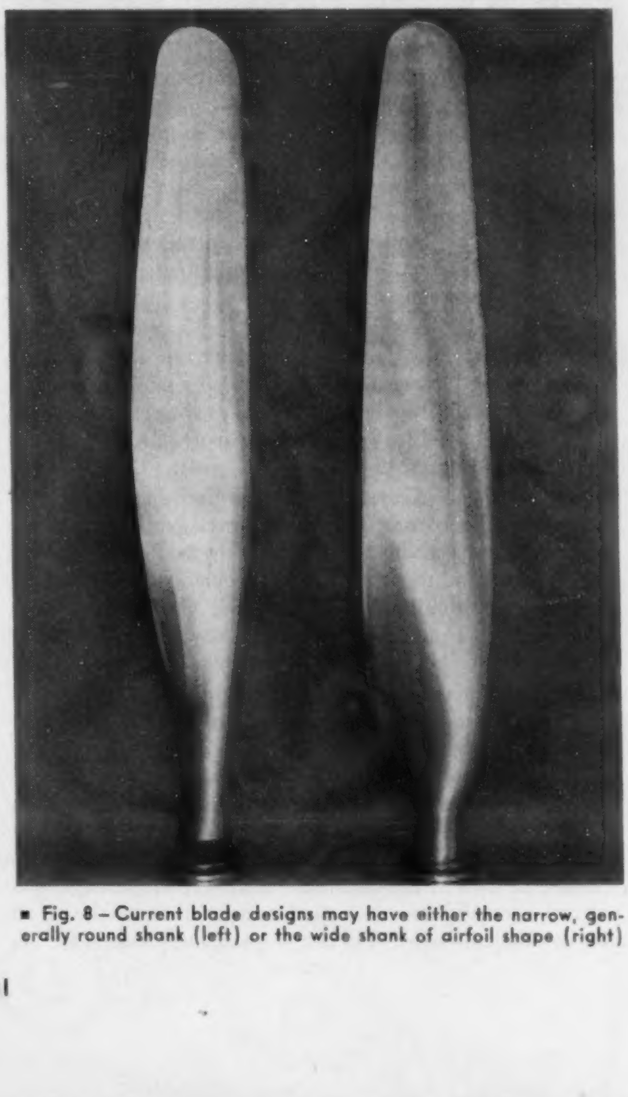
This process must be repeated for each important cruising condition and must be followed by a careful interpretation to give a propeller selection. The interpretation must include estimates for the other flight conditions, such as take-off, along with evaluation of propeller weight, size, and so on. Since the optimum selections for various flight conditions will usually be in conflict, it is important that the airplane designer know what conditions are to be emphasized. Where long range is involved, this requirement is particularly important, and may require extensive airplane design studies to evaluate a gain in cruising efficiency against, for example, a loss in single-engine performance.

A propeller selection, in cases of this sort, obviously cannot be made on propeller considerations alone, but may rather stem from the plans of the ultimate operator of the airplane. These plans should be part of the initial basic data for propeller selection, even though it is not always possible to place numerical values against the various items. If the day should ever arrive when a quantitative analysis of all of the factors can be made, this problem will be simplified. Such an analysis will require us to prove, for example, that a 1% change in cruising efficiency has the same value as 10 lb in weight, 2% in climb efficiency, 9 in. in diameter, and $100 in initial cost. We do not yet know how to derive the necessary equations to give such an answer.

A careful propeller selection analysis is ultimately desirable for every airplane to ensure the use of the best available equipment, but since an analysis to include every possible blade and hub can be burdensome, some considerations on the general type of propeller to be used will be helpful. It is often possible to decide in advance of detail studies whether the airplane should have single or dual rotation propellers, whether a three-blade propeller with wide blades or a four-blade propeller with narrow blades should be used. Other questions will cover the desirability of spinners, the need for blades with wide shanks for improved engine cooling, and the desirability of intentionally limiting propeller diameter beyond a point of good propeller performance in order to provide a compact airplane.

On the question of the best number of blades, it should be recalled that while the theoretically best propeller has an infinite number of infinitely narrow blades, the practical performance differences among propellers with varying numbers of blades providing the same total blade area are very small, and the effect on weight within the limits of present experience is usually small enough to be masked by other influences. In general, therefore, a three-blade propeller should be chosen instead of a four-blader when designs offering comparable blade areas are available. The three-blader will be cheaper to build, cheaper to maintain in service, and will be less likely to encounter vibration problems. A propeller having a greater number of blades than another is actually desirable only when the propeller operating forces that result from the use of excessively wide blades become too great to be handled in a practical manner. This point can best be determined by the manufacturer of the particular type of propeller under consideration. In the evaluation of this problem for dual rotation propellers, the same general considerations will apply, although the total number of blades involved will of course be larger.

In current blade designs, there is available, as indicated in Fig. 8, a general choice between those with narrow and



■ Fig. 8 – Current blade designs may have either the narrow, generally round shank (left) or the wide shank of airfoil shape (right)

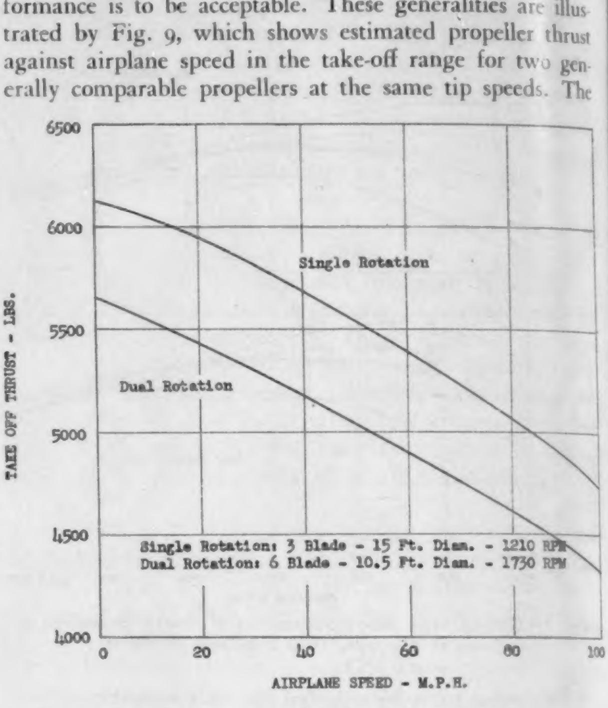
generally round shanks, and those with wide shanks of airfoil shape achieved either in the design of the blade or by the addition of shank fairings. The use of wide shanks is beneficial for cooling of aircooled engines under conditions of low airplane speed. Cooling for top speed or cruising is usually not affected to a measurable extent. The penalties involved in the use of wide shanks appear in the form of a greater propeller weight under otherwise equal conditions. The weight increase extends not only from the weight of the blade itself, but also from the weight of the operating mechanism necessary to handle the greater blade twisting moments. These twisting moments are markedly affected by the addition of weight to the leading and trailing edges of blade shanks. Apart from the cooling effects, there appears to be but little actual aerodynamic advantage in the wide shanks, particularly when used in front of radial engines. Where a choice is available, wide shanks that are integral with the propeller blade seem to be preferred over shank fairings, since for a given blade width, the airfoil thickness ratios for the latter will be larger. In general, however, wide shanks will give a heavier propeller than will round shanks with fairings.

The installation of propeller spinners is a subject that is currently more subject to speculation than to scientific analysis. Visual impression would always say that propeller spinners should be aerodynamically important. Experience, however, indicates that for tractor installation with radial aircooled engines, spinners actually have no marked effect on performance, and there are numerous cases on record where flight tests were unable to measure any effect at all on either airplane speed or engine cooling. In general, it appears the most that can be expected for improvement in top speed is about $1\frac{1}{2}\%$, and even this gain will not be realized except in a fortunate combination of circumstances. The small gains experienced to date indicate that the detail contour of the spinner is not very important. It is best, therefore, to make use of an existing design where possible, to avoid the extra burdens and delays involved in development of a new design. A reasonable selection of spinner sizes and shapes is now available to meet the usual requirements, if judgment indicates that a spinner must be used.

In spite of the present poor showing of spinners on radial installations, it seems fundamentally correct that spinners should be aerodynamically desirable, and it is probable that further research into spinner and cowling combinations will ultimately yield a configuration that will be considerably superior to present practice.

For liquid-cooled engines, the opinion is general that spinners must be used to provide fuselage lines that are adequate from the drag standpoint. It would be interesting, nevertheless, to see the results of comparative tests on an in-line installation in a high-speed airplane with the conventional spinner omitted, and the cowling lines behind the propeller faired to a reasonable contour.

Dual rotation propellers have passed through the development stage to a point where current propeller selection problems must include consideration of this type. The dual rotation propeller differs fundamentally from the single rotation of the same total blade area in only one major respect: Torque reaction and its associated aerodynamic losses are eliminated. The elimination of torque reaction in itself is important for relatively small airplanes, but it is normally not of so much consequence for the more heavily loaded types. The higher solidities that will inevitably be used with dual rotation propellers also imply that this type will be applied primarily to the relatively smaller airplanes, since high power loadings should be accompanied by large propeller disc areas if take-off performance is to be acceptable. These generalities are illustrated by Fig. 9, which shows estimated propeller thrust against airplane speed in the take-off range for two generally comparable propellers at the same tip speeds. The



■ Fig. 9 – Typical comparison of dual and single rotation propellers for 2000 take-off bhp at sea level

diameters of the propellers have been chosen so that they will operate in level flight at substantially the same blade airfoil lift coefficients, and will thus show efficiency differences in top speed only in so far as the basic type affects the efficiency. The results in take-off show clearly that if a propeller of large total solidity is to be used, it will be inferior in take-off to a propeller of lower solidity and suitably larger diameter.

The use of dual rotation propellers, in view of their practical relationship to propeller solidity, is important in connection with airplane designs that for one reason or another must use a propeller of restricted diameter. Diameter selection is, of course, one of the major problems in the entire propeller analysis, but it is far more than a propeller problem, in that the entire general arrangement of the airplane may be involved. From a study of current airplanes of all makes, it is apparent that opinions on the subject are diverse. In Fig. 1, a general range of expected propeller diameter was indicated, and normally the small diameters should be found on airplanes of the pursuit type or for low altitude, while the larger diameters should appear on heavy bombers or transports, and for high altitude. There are, however, existing propeller installations that fall in diameter below the low curve, even though the airplanes involved in some cases are not pursuit types.

It should be noted in this connection that some of those with very small propellers are, nevertheless, considered satisfactory. It appears impossible to provide any acceptable rules to cover this situation. Each airplane designer must satisfy himself on the diameter question in terms of his own airplane design figures. It is important only that in making his analysis, he should be fully aware of the pen-

alties involved when diameter is excessively restricted. When it must be restricted, he should endeavor to compensate as fully as possible for loss in diameter by obtaining a propeller of high solidity, even though solidity itself cannot fully regain the performance losses. It is pertinent to note in this connection that in general, propeller weight cannot be saved by restricting diameter, as the additional weight due to a suitable increase in solidity is of the same magnitude as the reduction due to decreased diameter. Airplane structural weight may, of course, be considerably affected by propeller size.

As a part of the propeller selection problem, the propeller installation requirements should be considered in a general way, and early in the detail design of the airplane the propeller installation should be thoroughly covered along with the other components of the powerplant. A review of the installation with the propeller manufacturer will be helpful, if undertaken before the details are frozen. For the Hamilton Standard feathering Hydromatic propeller, the installation will include three major parts: the propeller itself, the constant-speed control unit with its associated cockpit control, and the feathering pump with associated controls. Fig. 10 shows schematically a typical arrangement of propeller accessory equipment. This electric motor feathering pump system is actually only one of several possibilities, but it is the one generally used in current airplanes. *(See page 404.)*

The installation of the constant-speed control itself rarely offers severe problems to the airplane manufacturer. Mechanical interferences are most likely to be with reference to some part of the engine on which the control is mounted, and such interferences will normally have been corrected before a complete airplane installation is attempted. The airplane designer, therefore, sees two primary requirements: the provision of a cable control, or equivalent, from the cockpit to the constant-speed control shaft, and the installation of an oil line into the constant-speed control from the feathering pump. The primary requirements for the speed adjustment cable or rod are: freedom from excessive vibration, from lost motion, and from the effects of structural deflection. To meet these requirements a cable and pulley arrangement is often simplest although in some installations the routing of a suitable cable from the cockpit to the nose of the engine may be difficult. A lever and rod arrangement may then be found more convenient and many such installations have been successfully made. A rod arrangement should be designed to avoid excessive unsupported lengths at the propeller end of the system. Some troubles from this source have been experienced in the past in the form of accelerated wear on the constant-speed unit from the large forces associated with vibration of the rod.

For the more severe problems, an electrically actuated constant-speed control head is available in the form shown in Fig. 11. This unit replaces the mechanical head and pulley that is indicated in outline on Fig. 10. The installation requirements for this unit are relatively straightforward. The equipment will include a suitable single-pole, double-throw, momentary-contact switch for the cockpit, an indicator light to indicate maximum rpm setting of the control, a circuit breaker or equivalent for each control circuit, and a four-wire connection to each governor from its controls. *(See page 405.)*

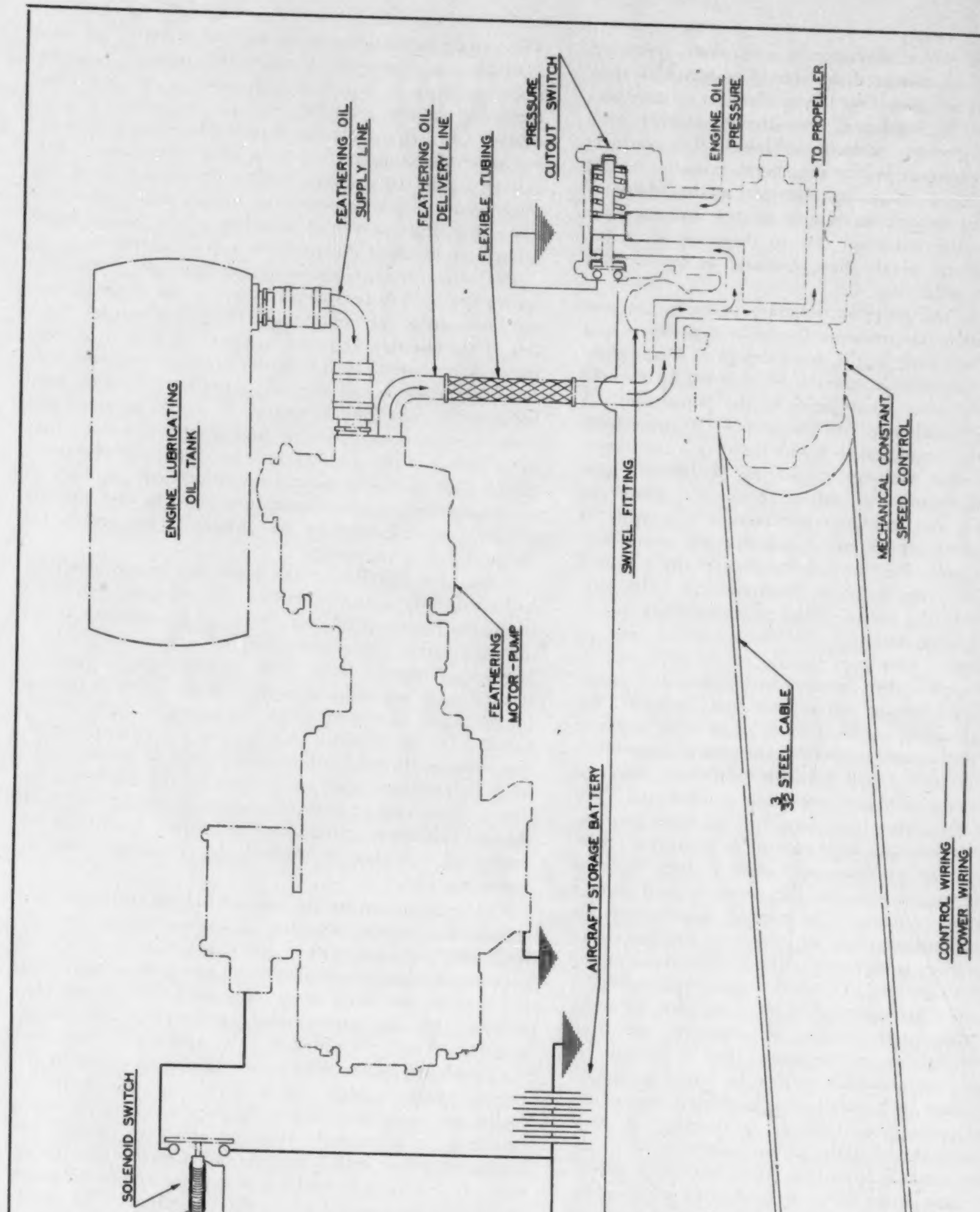
The feathering oil connection to the governor is a part of the emergency feathering system included in Fig. 10. This system has fundamentally but one objective: to make available to the propeller through the constant-speed control a quantity of liquid at high pressure for emergency feathering of the propeller, and for unfeathering if required. This requirement is, in principle, extremely simple, but adverse circumstances may in some cases make it difficult to fulfill. In common with other fluid systems, the combination of low atmospheric pressure and extremes of low temperature may lead to failure of the system unless it has been carefully designed for such conditions.

The principal requirement in the hydraulic installation for feathering is to provide a free flow of oil to the feathering pump under all conditions. The pump will not function if the pressure required to cause oil flow to it from the tank is more than is available from atmospheric pressure on the surface of the oil, plus the hydrostatic head between the tank and the pump. It should be noted that under adverse conditions this total available pressure may be as little as 3 psi. Feathering is an emergency operation; the oil lines do not in normal operation carry any flow of oil. Congealing at low temperatures must be prevented by a suitable arrangement of the plumbing, particularly for the inlet line to the pump.

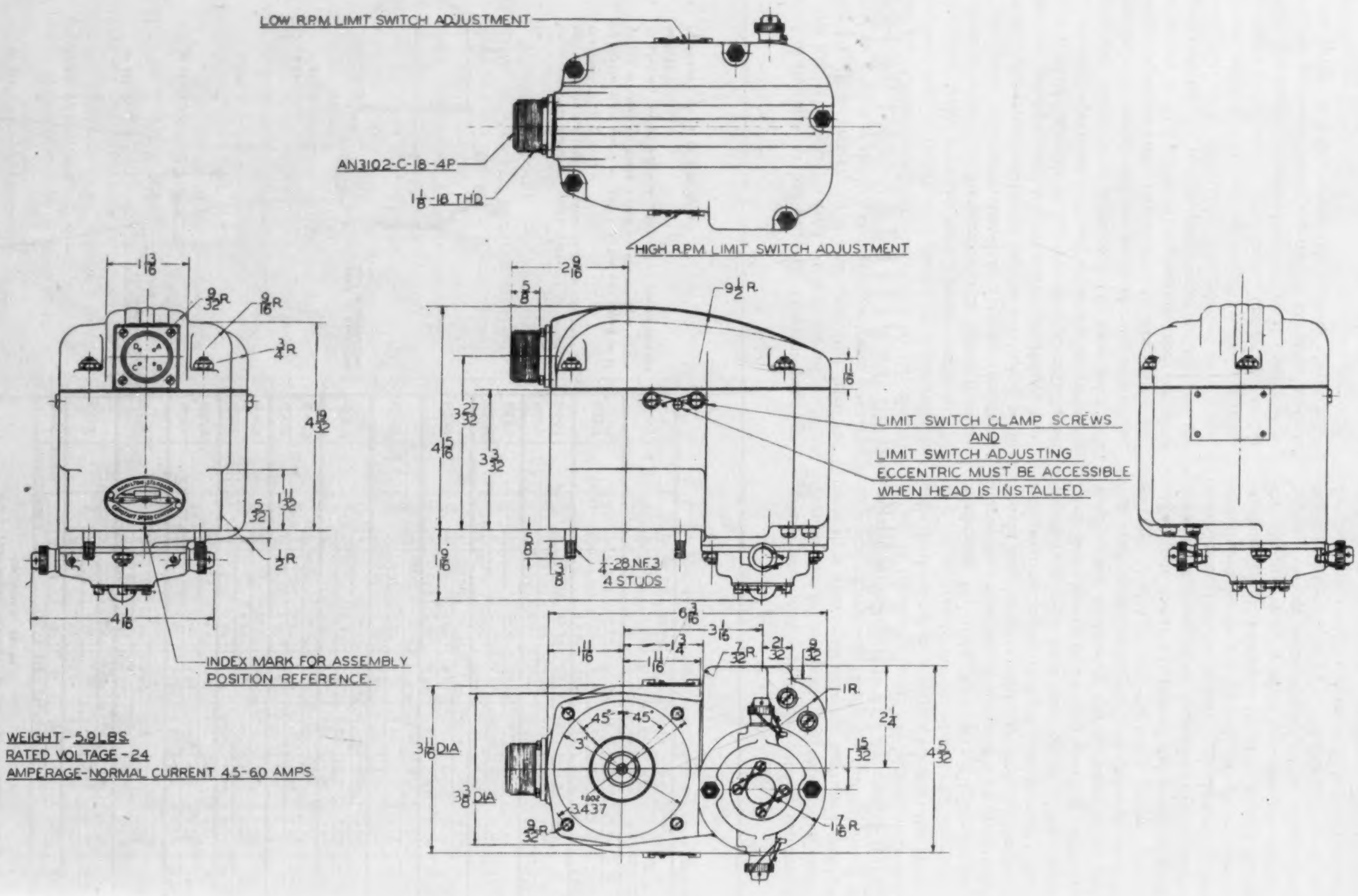
In an ideal installation, the feathering pump would be located directly at the outlet from the oil tank with but a negligible length of oil line. Such an arrangement is not always possible, but whenever an oil line to the pump is required, it should be as short and straight as possible. If the line is not short, it must be kept warm to prevent congealing of oil. In some cases, this may be automatically handled by the presence of a relatively high ambient air temperature. In other circumstances, special arrangements must be provided, such as location of the feathering oil line in direct contact with the engine oil supply line, with thermal insulation surrounding both pipes. Heat from the engine oil will then be available by conduction into the feathering line.

The arrangement for the inlet end of the feathering line should also receive attention. Common engine oil and feathering oil lines with a tee connection to divide oil between the engine and feathering pump have been used, but they are not likely to be successful under adverse conditions, and this arrangement is therefore not recommended. The principal difficulties appear to arise from the possibility of starvation of the feathering pump by the engine pump at high engine speeds, or conversely the feathering pump may starve the engine oil supply when feathering is attempted. It is preferable to provide the feathering pump with a separate outlet direct from the oil tank. It should be located adjacent to the engine oil outlet and should be provided with a short standpipe. The standpipe height should be less than that of the engine line to reserve a quantity of oil for feathering in case of general loss of oil from the engine system.

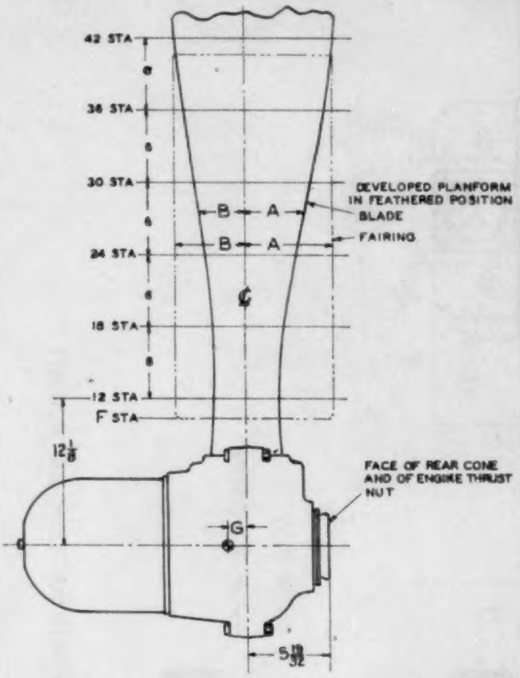
For the plumbing between the feathering pump and the constant-speed control, the requirements are somewhat less critical. If the feathering pump is properly fed, there will be available from it a sufficiently high pressure to move relatively cold and viscous oil into the propeller, although complete congealing of the oil will, of course, prevent feathering. For rapid feathering action, this line should be of generous size, and as free from bends as possible. Where the line runs between the fire wall and the engine, it will be kept warm by the surrounding air from the engine cylinders, and no lagging is necessary. For the short length



■ Fig. 10 – Diagrammatic sketch showing installation of Hydromatic accessories



■ Fig. 11 – Electrically actuated constant-speed control head



NOTES:

1. TO OBTAIN TRUE BLADE OR FAIRING WIDTH AT ANY STATION ADD DIMENSIONS A AND B LISTED IN TABLE FOR THAT STATION
2. TO OBTAIN BLADE ANGLE AT ANY STATION ADD VALUE OF Ø LISTED IN TABLE TO BLADE ANGLE AT REFERENCE STATION
3. F STATION DENOTES BLADE STATION AT WHICH FAIRING BEGINS
4. TOLERANCE ON G IS ± 1/16 INCH.

HAMILTON STANDARD PROPELLERS
HYDROMATIC PROPELLER INSTALLATION DIMENSIONS
HUB MODEL 23E50

BLADE	FAIRING MODEL	DIMEN. G		BLADE STATIONS 9	12	18	24	30	36	42	72	F
6339		1	A		2.62	3.26	4.16	4.96	5.34	5.16		
			B		2.38	2.74	3.34	3.88	4.16	3.98		
			ø		24.5°	17.4°	11.0°	5.6°	2.25°	0°		
6339	A	1	A		5.94	5.94	5.89	5.75	5.53	5.16		12
			B		4.66	4.68	4.65	4.57	4.35	3.98		
			ø		24.5°	17.4°	11.0°	5.6°	2.25°	0°		
6353 6429		1	A		2.60	3.68	4.88	5.74	6.00	5.95		
			B		2.40	3.04	3.90	4.52	4.74	4.69		
			ø		24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6353 6429	B	15/16	A		3.30	5.55	6.36	6.36	6.06	5.95		12
			B		5.09	5.17	5.16	5.09	4.80	4.69		
			ø		24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6353 6429	C, H	15/16	A	* 6.91	6.81	6.67	6.50	6.31	6.18	5.95		8
			B	* 5.29	5.27	5.23	5.15	5.04	4.92	4.69		
			ø	* 27.1°	24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6359		7/8	A		2.40	2.96	4.01	5.05	5.90	6.50		
			B		2.40	2.74	3.39	4.11	4.68	5.10		
			ø		33.14°	23.64°	15.94°	9.74°	3.34°	0°		
6359	A, L	13/16	A	5.20	5.18	5.88	6.93	7.35	7.30	7.08		9
			B	5.05	5.25	5.57	5.78	5.81	5.71	5.57		
			ø	38.3°	33.14°	23.64°	15.94°	9.74°	3.34°	0°		
6359	C	13/16	A	5.21	5.31	5.55	5.76	5.92	5.96	6.50		9
			B	4.81	4.81	4.81	4.81	4.81	4.74	5.10		
			ø	23.24°	20.94°	16.44°	12.14°	7.64°	3.34°	0°		
6379 6447		1 1/8	A		2.61	3.25	4.16	4.96	5.34	5.12		
			B		2.39	2.75	3.34	3.88	4.16	3.98		
			ø		27.3°	20.2°	13.8°	8.4°	3.8°	0°		
6379 6447	A	1 1/16	A	5.39	5.41	5.49	5.33	5.02	5.34	5.12		9
			B	4.60	4.58	4.50	4.32	3.94	4.16	3.98		
			ø	30.85°	27.3°	20.2°	13.8°	8.4°	3.8°	0°		
6393		1	A		2.55	3.63	4.83	5.69	5.95	5.90		
			B		2.45	3.09	3.95	4.57	4.79	4.74		
			ø		28.65°	21.55°	15.15°	9.45°	4.4°	0°		
6393	A, B	1	A	5.10	5.19	5.40	5.59	5.76	5.95	5.90		9
			B	4.60	4.60	4.60	4.60	4.68	4.79	4.74		
			ø	32.25°	28.65°	21.55°	15.15°	9.45°	4.4°	0°		

* THESE DIMENSIONS ARE AT THE 8 INCH STATION

forward of the cylinder baffles, to the constant-speed control, the line is exposed to cold air and should be thoroughly lagged to preserve in it heat conducted from the warmer aft portion of the line and from the warm constant-speed control.

The above recommendations are based on the general assumption that all installations should be made suitable for operation under even the most adverse conditions. In any airplane where it is known that only relatively favorable conditions will be involved in routine operations, many of these recommendations can be overlooked without serious consequences. Under today's conditions, however, it is rarely possible to be sure that an installation based on such optimism will not ultimately be flown in theaters very remote from those originally intended.

The electrical installation for the feathering system can be relatively simple and straightforward. The wiring diagram shown in Fig. 10 is self-explanatory and the space requirements for the cockpit push-button switch and the solenoid relay are a minimum. The solenoid switch may be located as desired for wiring simplicity, and for minimum length of the power circuit lines. The cockpit switch location is a matter of individual choice. It should be so chosen that accidental feathering will be unlikely, but such that the switch is within reasonable reaching distance for the pilot.

This question of switch location is subject to many opinions. Some operators feel that the feathering controls should intentionally be made relatively inaccessible, on the ground that more trouble can result from feathering too soon than from small delays in the time required to feather in a real emergency. This is a real likelihood in a four-engined airplane where it is not easy to tell at a glance which of the four engines has become inoperative. If the pilot feathers before a correct selection has been made, he may be reduced to the use of two engines, where without feathering he could have retained three. Location of feathering switches in accordance with this philosophy should not, however, make selection of the proper switch or operation of the switches difficult, once the necessary preliminary steps have been taken. These steps may include the releasing of a switch guard, which may serve the dual purpose of delaying the feathering action and preventing inadvertent feathering by accidental contact with the switch.

Provision for installation of the propeller itself raises a question of dimensional requirements. Clearance for the propeller diameter is, of course, fundamental. Provision of clearance between the propeller and the adjacent engine cowling is not difficult if properly studied fairly early in the airplane design, when aerodynamic studies for the engine cowling arrangement are in progress. Propeller outline charts similar to that shown as Fig. 12 are available for the various combinations of hubs and blades, with and without shank fairings. In some cases, the provision of the desired cowl outline may conflict with the location of

HAMILTON STANDARD PROPELLERS
HYDROMATIC PROPELLER INSTALLATION DIMENSIONS
HUB MODEL 23E50

BLADE	FAIRING MODEL	DIMEN. G		BLADE STATIONS 6	12	18	24	30	36	42	72	F
6435		1	A		2.60	3.68	4.88	5.74	6.00	6.01		
			B		2.40	3.04	3.90	4.52	4.74	4.81		
			ϕ		24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6435	B	15/16	A		3.30	5.55	6.36	6.36	6.06	6.01		12
			B		5.09	5.17	5.16	5.09	4.80	4.81		
			ϕ		24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6435	C, H	15/16	A	* 6.91	6.81	6.67	6.50	6.31	6.18	6.01		8
			B	* 5.29	5.27	5.23	5.15	5.04	4.92	4.81		
			ϕ	* 27.1°	24.5°	17.4°	11.0°	5.6°	2.16°	0°		
6443		13/16	A		2.40	2.96	4.01	5.05	5.90	6.50		
			B		2.40	2.74	3.39	4.11	4.68	5.10		
			ϕ		45.55°	36.05°	28.35°	22.15°	15.75°	12.41°	0°	
6443	A	3/4	A	5.20	5.18	5.88	6.93	7.35	7.30	7.08		9
			B	5.05	5.25	5.57	5.78	5.81	5.71	5.57		
			ϕ	50.71°	45.55°	36.05°	28.35°	22.15°	15.75°	12.41°	0°	
6443	C	3/4	A	5.21	5.31	5.55	5.76	5.92	5.96	6.50		9
			B	4.81	4.81	4.81	4.81	4.81	4.74	5.10		
			ϕ	35.65°	33.35°	28.85°	24.45°	20.05°	15.75°	12.41°	0°	
6443	E, J, L	3/4	A	5.20	5.18	5.88	6.93	1.35	7.30	7.08		9
			B	5.05	5.25	5.57	5.78	5.81	5.71	5.57		
			ϕ	50.71°	45.55°	36.05°	28.35°	22.15°	15.75°	12.41°	0°	
6477		7/8	A	※ 2.62	3.96	5.81	6.52	6.88	7.02	6.93		
			B	※ 2.62	3.34	4.13	5.29	5.56	5.67	5.60		
			ϕ	※ 21.6°	24.8°	16.5°	12.0°	5.6°	2.26°	0°		
6491		13/16	A	※ 2.62	3.86	5.81	6.48	6.64	6.62	6.53		
			B	※ 2.62	3.34	4.73	5.43	5.83	6.02	6.10		
			ϕ	※ 36.9°	35.3°	30.3°	25.7°	21.2°	17.1°	13.5°	0°	
6501 6507		13/16	A	※ 2.62	3.86	5.81	6.48	6.64	6.62	6.53		
			B	※ 2.62	3.34	4.73	5.43	5.83	6.02	6.10		
			ϕ	※ 23.4°	21.8°	16.8°	12.2°	7.7°	3.6°	0°		
			A									
			B									
			ϕ									
			A									
			B									
			ϕ									

* THESE DIMENSIONS ARE AT THE 8 INCH STATION
※ THESE DIMENSIONS ARE AT THE 10 INCH STATION

DATA SHEET-178

■ Fig. 12 – Propeller outline chart

the propeller blade trailing edge in the feathered condition. The solution of the problem is, however, one of simple arithmetic, and should at most have no more serious effects than to introduce into the original propeller selection an additional element for consideration. In the choice between blades with wide and narrow shanks, clearances should be considered, particularly when the installation involves a relatively short coupled engine where provision of a good cowl nose between the maximum engine diameter at the cylinders and the propeller blade trailing edge is difficult.

The installation of non-feathering propellers for single-engine airplanes differs in the case of the Hydromatic by omission of the feathering pump, with its controls, wiring, and plumbing. The propeller itself, the constant-speed control unit, and the cockpit control are normally all that are required. The problems encountered are analogous to those for any normal control linkage and need no special precautions. Other Hamilton standard designs, the constant-speed counterweight type and the two-position controllable, are equally simple in installation requirements, and involve nothing that is not covered by the analogous equipment for the Hydromatic.

■ ■

Aircraft Oil Systems—High-Altitude Problems

continued from page 396

a stop that is adjusted to keep the two off-center holes aligned. Scavenge oil enters at the top, rotates around the cylinder, forming a vortex, and leaves at the bottom. As it enters the separator and forms a vortex, the entrained air is released from the oil into the air space above the vortex and, under scavenge pressure, is forced out through the air vent. The surface of the vortex then rises until its edges touch the rotor blades. The rotor, being very lightly spring-loaded, is turned and the air exit closes. No more air can escape and the surface of the vortex drops again causing the hole to open. In actual operation the vortex seeks a fairly constant level depending upon the amount of air entrained in the oil. Effective separation is obtained with this unit down to approximately 5% entrained air. However, 5% entrained air represents a loss of 6% to 7% of oil pump output, and further reduction is desirable.

Satisfactory operation of the Sharples separator is dependent upon maintaining the vortex. Maneuvers, such as inverted flight or sudden negative acceleration, destroy the vortex, temporarily stopping air separation and allowing oil to escape through the air vent. Further development may improve the effectiveness and adaptability.

In conclusion, it appears that satisfactory lubrication system design has become an item requiring serious consideration. We can summarize the problem by saying that temporary corrections are available immediately, but most of them entail certain disadvantages and require some flight research. Complete corrections are neither complex nor unreasonable but cannot be immediately available without disadvantages of their own.

Errata

We regret that the illustrations accompanying Figs. 1 and 2 of R. L. McBrien's paper "An Aircraft Double Windshield – Its Development and Use" were reversed. Referring to page 351 of the October issue, the picture shown above Fig. 2 should accompany Fig. 1 and the picture above Fig. 1 should accompany Fig. 2.

THE INFLUENCE of DIESEL FUE
on ENGINE DEPOSIT

by G. H. CLOUD and A. J. BLACKWOOD

Esso Laboratories, Standard Oil Development Co.

THE introduction of the high-speed, low-weight-per-horsepower compression-ignition engine into the automotive field was soon followed by the realization that such equipment may be rather sensitive to changes in fuel properties. Considerable work has been done and reported in the literature in studies of these engines relating the various fuel properties to engine power, fuel consumption, roughness, smoke, and odor. However, very little authoritative work has been reported on the effect of fuel characteristics on engine fouling and wear. The belief has been almost universal that diesel-engine degradation is predominantly affected by lubricating oil, and that practically any clean fuel that can be made to burn smoothly will provide satisfactory operation.

Some work on automotive diesels, reported in 1938 by MacGregor and Hanley[1] indicated that under special conditions certain fuel characteristics affected carbon formation in the combustion chamber. Also, work on larger and essentially stationary engines by Broeze and Gravesteyn,[2] reported in 1938, indicated that sulfur content of diesel fuels affected engine degradation if it was present in amounts greater than 1% by weight of the fuel. The CFR report for 1941[3] on fuel performance in full-scale automotive diesel engines indicated that fuel characteristics may affect combustion-chamber carbon deposits under the abnormal conditions of extremely low intake air pressures. These reported values, although not particularly conclusive as applying to routine operation of automotive diesels, did indicate a definite effect of fuel characteristics, and when it became evident from observations of automotive diesel equipment in service that lubricant characteristics could not per se account for the differences in the observed fouling and wear, the Esso Laboratories initiated a detailed and specific study of the effects of fuel characteristics on automotive diesel-engine condition. This paper is a summary of the results of this investigation.

THE introduction of the high-speed, low-weight-per-horsepower diesel engine brought the realization that such equipment might be rather sensitive to changes in fuel properties. Observations of automotive diesel equipment in service showed that lubricant characteristics could not in themselves account for the differences in the observed fouling and wear. These reports led the authors to initiate a series of tests designed to study the problem specifically and in more detail.

From these studies, it became clear that the selection of suitable fuels is an important factor in the control of fouling and wear in automotive diesel engines. Assuming stability, freedom from water and soaps, and absence of corrosive abrasive, and residual materials, the major diesel fuel characteristics governing automotive diesel-engine fouling and wear are sulfur content, ignition quality, viscosity, and/or volatility.

Of these properties, sulfur content is the most important. Increasing the sulfur content from 0.2% to 1.0% may result in a 40 to 80% increased engine fouling and a two to sixfold increase in ring and cylinder wear.

Using detergent types of lubricants will only partially alleviate the fouling and wear due to the fuel.

The influence of these fuel properties, as determined by the authors, particularly the influence of sulfur on fouling and wear, apparently cannot be applied directly to larger slow-speed diesels in marine and industrial service.

• • •

THE AUTHORS: A. J. BLACKWOOD (M '27) and G. H. CLOUD (M '38), as an author team are well known to SAE members. Their last paper in the *SAE Journal*, "Characteristics of Diesel Fuels Influencing Power and Economy," appeared in October, 1940. Mr. Blackwood, who is in charge of Standard Oil Development Co.'s engine laboratories, and of testing and development of automotive, diesel and aviation fuel, also contributed and collaborated on earlier papers on diesel fuels and detonation. His colleague, Dr. Cloud, heads Esso Laboratories' diesel-fuels research work.

■ Effect of Load on Fouling and Wear

Prior to discussing the effect of fuel characteristics, it may be well to indicate briefly the effect of engine load on the engine condition items to be discussed, for one of the three engines used in the investigation, when operating on the normal automotive diesel fuel used as a reference fuel. This should serve to permit the comparison of data on

[This paper was presented at the SAE Diesel-Engine and Fuels & Lubricants Meeting, Cleveland, Ohio, June 2, 1943.]

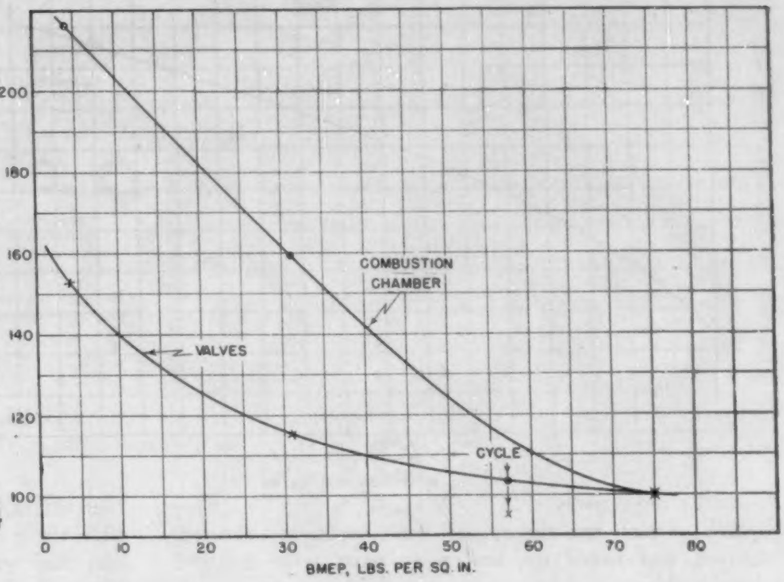
[1] See SAE Transactions, Vol. 43, July, 1938, pp. 272-280: "Diesel Deposits as Influenced by Fuels and Operating Conditions," by J. R. MacGregor and W. V. Hanley.

[2] See *British Motor Ship*, Vol. 19, September, 1938, pp. 216-218: "Fuel and Wear in Diesel Engines," by J. J. Broeze and B. J. J. Gravesteyn.

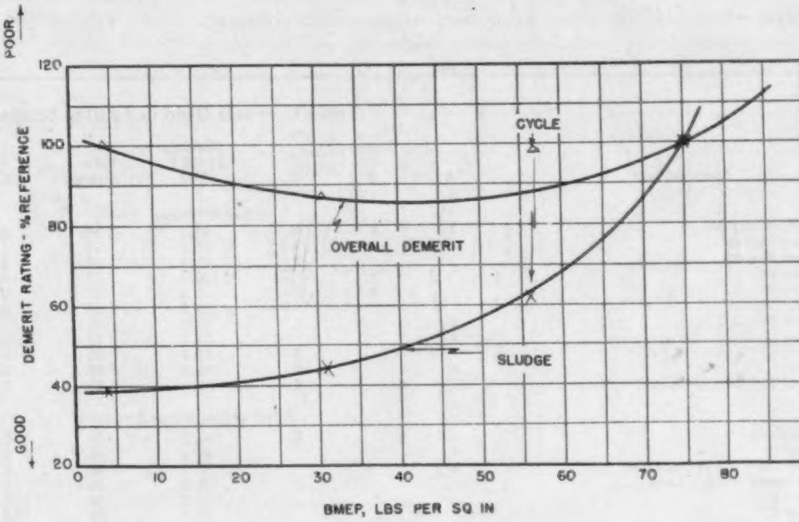
[3] See SAE Transactions, Vol. 49, October, 1941, pp. 448-460: "Evaluation of Diesel Fuels, in Full-Scale Engines" (CFR Report), by W. G. Ainsley.

E ROPERTIES
T nd WEAR

other engines and other conditions of which the reader may have knowledge, and should, in general, be helpful in providing a starting point for the development of the subsequent discussions. In our laboratories, engine condition is evaluated by a point system of demerits charged against the part of the engine being examined, and a weighted overall demerit is calculated in cases where the engine as a whole is rated. Values are expressed then as a per cent of reference, with values above 100% representing a poorer condition than with the reference fuel. Values below 100% are better than with the reference fuel.



■ **Fig. 1 – Effect of load on combustion-chamber and valve condition – engine A, 80-hr tests at 1600 rpm**



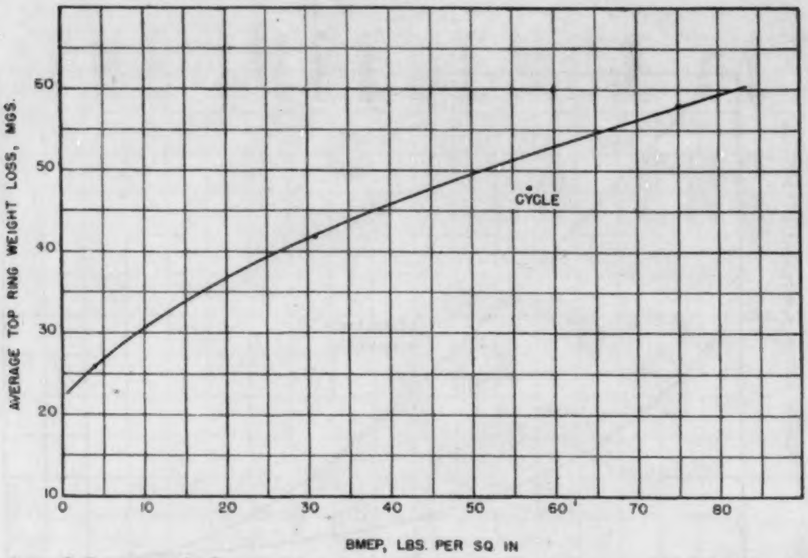
■ **Fig. 2 – Effect of overall demerit rating and sludge formation – engine A, 80-hr tests at 1600 rpm**

The tests shown graphically in Figs. 1, 2, and 3 were made on the reference fuel, and 75 bmep was used as the reference point. The effect of engine load on combustion-chamber and valve condition is illustrated in Fig. 1 for 1600 rpm. Both combustion-chamber and valve condition are shown to improve with increasing engine load. For the sake of brevity, the effect of speed is not shown but, in general, an increase in speed accelerates the accumulation at light loads. In cycle operation where the engine was idled for 3 min and run at about full load for 7 min, the fouling based on the average bmep corresponded quite closely to that predicted from the constant-load curve.

The effect of load on overall demerit rating and sludge formation is shown in Fig. 2. Although combustion-chamber fouling is markedly greater at light load than at full load, ring sticking and lubricating oil sludge formation increase with engine load to such an extent that they offset carbon formation and show an overall demerit that changes little with load up to 75 bmep, which is rated load for engine A. As illustrated in Fig. 3, engine wear as represented by top-ring weight loss increases with engine load. Again, the averaged bmep for cycle operation is in line with results obtained at constant speed.

■ Engines and Conditions Selected for Test

After a careful weighing of the various factors involved, it was ultimately decided to conduct a series of 80-hr tests on three different makes of automotive engines, operating two of them in cyclic fashion broadly to simulate field service. Accordingly, engines A and C were run on a 3-min idle and 7 min at 90% rated load cycle. Engine B was run constant speed and 120% rated load, which represented conditions reported to be giving trouble in the field on this type of engine. A total of about 80 tests covering over 40 fuels was run on these three engines, and the results from them represent the body of the data presented here. In addition, the observations made on these three engines have been confirmed by tests on two other types of engines in the laboratory, and have been substantiated by extensive service tests on one engine type in the field. A commercially accepted compounded lubricant of the detergent type was used in the crankcase of engine B and straight paraffinic and naphthenic lubricants were used in engines



■ Fig. 3 – Effect of load on top-ring weight loss – engine A, 80-hr tests at 1600 rpm

A and C, respectively.

Prior to each test the engine was thoroughly cleaned, adjusted, and tuned up, and worn parts were replaced where necessary to ensure against engine changes causing differences in the test results. In studying the effects of a given fuel characteristic, the test fuels were selected in groups and in such a way as to change one variable at a time while holding other properties substantially constant.

Table 1 gives the pertinent inspection analyses of the fuels used with key columns indicating which fuels were run in the different engines. The table is arranged so that, in general, fuels selected as representing, for example, a series of varying cetane number but constant viscosity, are in a group. Table 2 gives a broad description of the three engines used in this investigation.

■ Effect of Sulfur on Engine Cleanliness

From observations made quite early in the course of the program, it became evident that the sulfur content of the fuel might be a major factor affecting engine condition from the standpoint of both fouling and wear. This factor was then studied quite exhaustively by selecting fuels of varying sulfur content and by the preparation of special blends where the type and amount of sulfur were controlled. By and large, as will later be shown, the magnitude of the influence of the various other fuel characteristics on engine condition is not of such proportions as to give

Table 1 – Fuels Used in Fouling Studies*

Fuels Tested	Engine A	Engine B	Engine C	Viscosity at 100 F, SUS	Cetane Number	Sulfur, %	ASTM Distillation 10%	ASTM Distillation 50%	ASTM Distillation 90%	ASTM Distillation FBP
Effect of Ignition Quality										
Reference Fuel	X	X	X	35.0	54	0.23	440	503	580	642
Test Fuel 28	..	..	X	35.2	36	0.23	416	486	568	636
Test Fuel 29	..	..	X	35.2	44	0.23	428	496	572	640
Fuel 30	..	X	..	36.8	32	0.25	438	448	582	651
Fuel 35	..	X	..	35.4	37	0.24	424	496	578	660
Fuel 36	..	X	..	35.5	46	0.23	430	496	578	628
Fuel 38	X	..	..	35.6	38	0.25	420	492	564	606
Fuel 12	X	..	..	35.3	50	0.20	444	500	593	638
Fuel 40	..	..	X	35.9	65	0.27	440	516	620	668
Fuel 2	X	..	X	33.0	38	0.82	406	466	550	610
Fuel 2 + 2.5% Isobutyl Nitrate	..	..	X	33.0	51	0.82	408	468	548	604
Effect of Viscosity and Volatility										
Kerosene	X	..	X	30.2	49	0.10	372	430	502	544
Fuel 10	X	..	..	35.3	49	0.15	460	490	538	586
Fuel 10 + 3% Heavy Lube	X	..	..	35.5	50	0.14	462	490	550	674
Kerosene + Light Lube	..	..	X	34.9	54	0.23	384	474	694	754
Fuel 26	..	X	..	35.4	50	0.41	448	504	576	648
Fuel 27	..	X	..	66.0	52	0.39	572	650	720	cracked
Fuel 34	..	..	X	35.4	52	0.53	442	508	586	640
Fuel 37	..	..	X	39.6	54	0.52	470	556	665	710
Fuel 39	..	..	X	44.5	56	0.52	498	600	692	726
Fuel 42	..	..	X	32.3	50	0.54	388	458	545	626
Reference Fuel	X	X	X	35.0	54	0.23	440	503	580	642
75:25 Reference Fuel + Heavy Gas Oil	X	..	..	37.6	55	0.30	448	528	638	696
50:50 Reference Fuel + Heavy Gas Oil	X	..	..	40.3	56	0.38	468	564	670	716
Heavy Gas Oils	X	..	..	51.0	58	0.53	536	630	690	720
Effect of Sulfur										
Fuel 4	X	..	X	33.4	44	0.69	406	482	566	622
Fuel 4 + 1% Diamyl Trisulfide	..	..	X	33.4	51	1.10	402	476	566	624
Fuel 6	X	..	..	34.3	48	0.46	422	492	570	631
Fuel 7	..	X	..	33.4	44	0.68	414	476	558	614
West Texas Gas Oil (unsweetened)	..	..	X	35.4	48	0.82	440	504	582	652
West Texas Gas Oil (sweetened)	X	X	X	35.4	57	0.96	446	506	578	614
West Texas Gas Oil (sweetened) + Nitrogen Tetrasulfide	..	..	X	35.3	57	1.00	448	506	576	614
West Texas Gas Oil (soda washed)	..	..	X	35.8	50	1.10	444	510	594	632
West Texas Gas Oil (sweetened)	..	..	X	35.9	51	1.20	445	512	594	634
West Texas Gas Oil (sweetened) – 98% Overhead	..	..	X	35.1	47	1.05	442	502	588	618
Gulf Coast Gas Oil	..	X	..	49.9	44	1.35	524	610	710	754
Paraffinic Gas Oil	..	..	X	40.0	55	0.60	454	568	666	710
Reference Fuel + 0.6% CS_2	..	X	..	35.0	54	0.95	440	503	580	642
High Sulfur Gas Oil	..	..	X	36.0	50	1.32	432	502	586	664
Reference Fuel + SO_2 (intake air)	..	X	..	35.0	54	0.56	440	503	380	642
Hydrogenated Gas Oil	..	..	X	35.4	54	0.06	412	506	618	670
Narrow Cut Gas Oil	X	..	X	34.4	50	0.23	470	489	522	545

* This tabulation presents only a general classification of the fuels. Data from almost all fuels tested in a given engine were used in determining the effect of the various fuel characteristics.

Table 2 – Engines Used in Fouling Tests*

Characteristic	Make of Engine: Hercules	General Motors	Caterpillar
Model	DJXBL	3-71	Experimental
Type	4-stroke cycle	2-stroke cycle	4-stroke cycle
Number of cylinders	6	3	1
Bore, in.	3½	4¼	5¾
Stroke, in.	4½	5	8
Displacement, cu in.	260	213	207.5
Compression Ratio	15	16	16
Rated Bhp	77 at 2600 rpm	83 at 2000 rpm	16.7 at 850 rpm
Rated Bmep	102	70	75
Injection system	Jerk pump (Bosch)	Unit injectors (own make)	Jerk pump (own make)
Combustion Chamber	Ante-chamber turbulent-type	Open chamber direct injection	Precombustion type

* Each engine was connected directly to an electric dynamometer and equipped with the necessary thermocouples to permit proper maintenance of specified operating conditions.

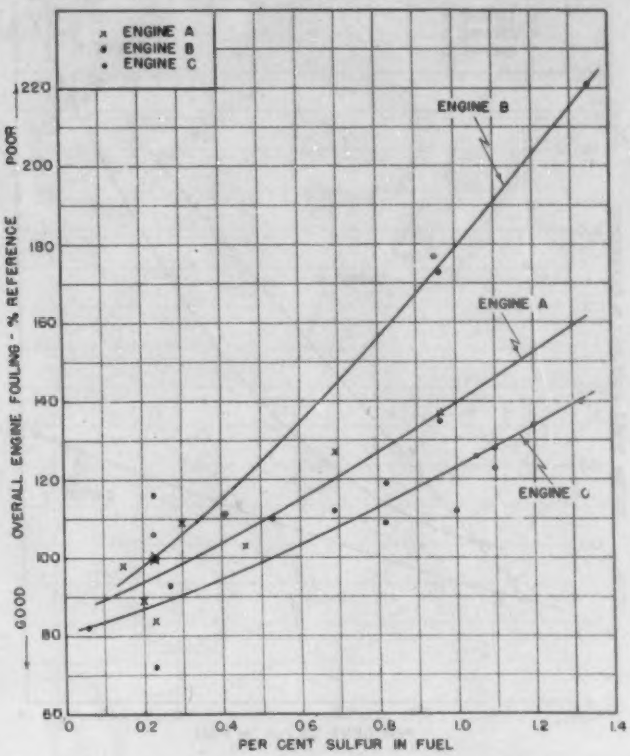
a fuel supplier any serious concern when considering existing limits of automotive diesel fuel specifications in the different grades.

The effect of sulfur on engine fouling is illustrated in Fig. 4, which shows the relation between per cent sulfur in the fuel and engine fouling as measured by overall demerit rating. A change in sulfur content from about 0.2% to 1% resulted in fouling increases of approximately 40, 50, and 80%, respectively, for engines C, A, and B. The high degree of fouling in engine B with high sulfur fuel was no doubt due to the fact that this engine was run at high load where blowby is greatest and where the greatest amount of fuel is burned per test.

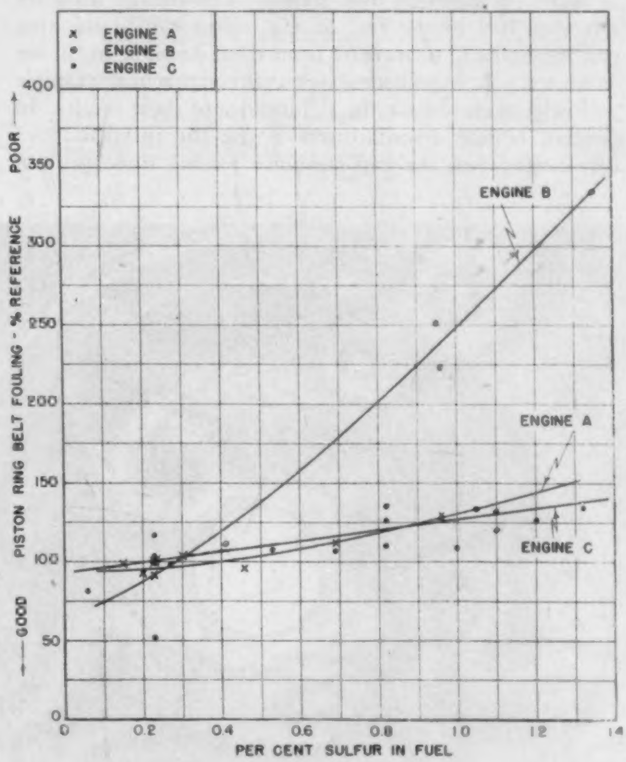
The effect of sulfur content of the fuel on ring-zone condition is shown in Fig. 5. Here again engine B shows markedly more fouling as sulfur is increased, and operation on the high sulfur fuel caused severe sticking of the two top compression rings and a heavy varnish deposition on the rings, lands, and grooves. This engine operating on the low sulfur reference fuel was relatively clean, with complete freedom from ring sticking.

It is interesting to note that varnish deposition on the piston skirts increased with increasing sulfur content. This effect is illustrated for all three engines in Fig. 6, showing the relation between sulfur content and piston-skirt fouling. Engine A appeared to be the most sensitive to changes in sulfur content with respect to piston-skirt fouling. Although engine C was least sensitive to piston-skirt varnish formation, it showed a peculiar response to increase in sulfur content of the fuel in that the high sulfur fuels formed varnish on the cylinder walls in the path of ring travel. This may be due to some influence of surface temperature which influences where the varnish deposits will settle.

Fig. 7 shows the effect of sulfur on the valve seats of engine C. The high sulfur fuels showed a general tendency to foul valves and stems and corrode seats, but these phenomena were quite marked in engine C. After 80 hr of operation on a fuel with a sulfur content of about 1% or above, the valves and seats were pitted to such an extent that it was necessary to reface them before continuing test work.



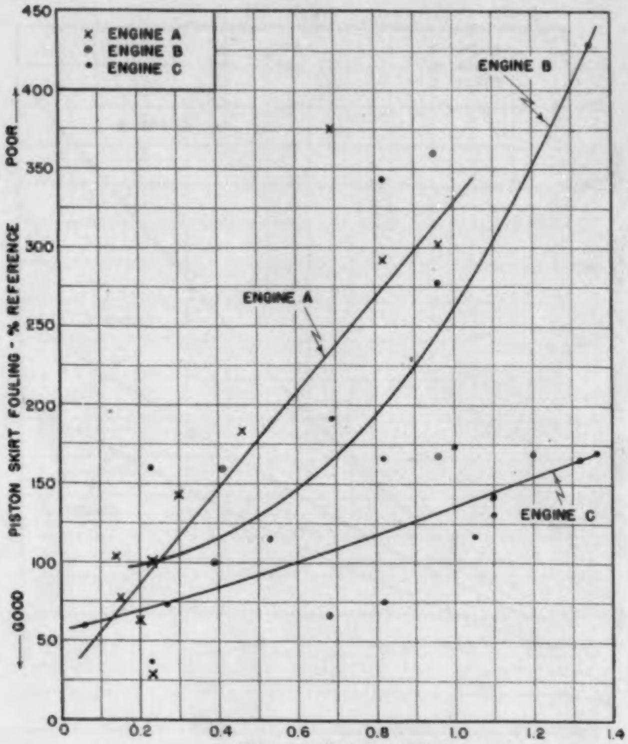
■ Fig. 4 – Effect of sulfur content of fuel on overall engine fouling – 80-hr tests



■ Fig. 5 – Effect of sulfur content of fuel on piston-ring-belt fouling – 80-hr tests

■ Effect of Sulfur on Engine Wear

In addition to increasing engine fouling, an increase in sulfur content of the fuel results in greater engine wear, and the results show sulfur content to be the most important fuel characteristic affecting engine wear. The relation between top-piston-ring wear and per cent sulfur in the fuel is shown in Fig. 8. In changing from the reference fuel of 0.23% sulfur to a fuel of 1% sulfur, ring wear in engine B increased almost threefold, while in engines



■ Fig. 6 – Effect of sulfur content of fuel on piston-skirt fouling – 80-hr tests

A and C it increased over sixfold. The change from the reference fuel to the fuel of 1% sulfur caused the ring gap in engine C to increase from 0.002 to 0.014 in. in the 80-hr test. A hypothetical yet entirely practical example will help to emphasize the significance of these results. In general, engine manufacturers advise the installation of new rings when the gap clearance reaches 0.060 in. On this basis, if engine C were installed in a truck chassis, it would be possible to operate the unit on the reference fuel (0.23% S) approximately 32,000 miles without reringing, whereas the use of a fuel of 1% sulfur would necessitate the installation of new top rings after only 5000 miles of operation.

The relation of cylinder wear to fuel sulfur content is shown in Fig. 9. Again comparing the results of changing from the reference fuel to a fuel of 1% sulfur, cylinder-liner wear on engine A showed a fourfold increase, and on engines B and C was more than doubled. It is interesting to note that the effect of sulfur on engine condition varies from unit to unit. Engine A is most sensitive with respect to cylinder wear; engine B to overall fouling; and engine C to ring wear.

■ Action of Sulfur in Fouling

As soon as it had been established that sulfur content of diesel fuels markedly affects engine fouling and wear, a study was initiated to determine the role played by the sulfur in these processes. The results obtained have shown the type of sulfur present in the fuel to be relatively unimportant. Blends of various sulfur compounds in the reference fuel, such as carbon disulfide and diamyl trisulfide, resulted in engine fouling and wear of the same order as that obtained on conventional fuels of similar characteristics and the same sulfur content. Also, the addition of small quantities of sulfur dioxide to the intake air of the diesel engine increased fouling and wear. These results indicated that the products of combustion of the sulfur in fuels, and not the type of sulfur compound present, were responsible for the effects observed.

If this conclusion could be simply demonstrated to be a fact beyond question, much of the mystery regarding the influence of type and amount of sulfur in the fuel might be alleviated. Accordingly, a study of diesel exhaust gases was initiated. Very careful analyses of exhaust gases

■ Fig. 7 – Valve-seat corrosion – 80-hr test on fuel of 1% sulfur – engine C

from a variety of diesel engines showed that under normal conditions 60 to 90% of the sulfur present in the fuel is converted to sulfur trioxide during the combustion process. This is contradictory to the popular belief, and to statements appearing in the literature that the sulfur burns to sulfur dioxide. In order to determine the relative effects of sulfur dioxide and sulfur trioxide on engine fouling and wear, tests were run in a CFR engine in which the sulfur dioxide was introduced into the intake air under both motoring and firing conditions; and sulfur trioxide was introduced into the intake air under motoring conditions. The results showed the sulfur dioxide to have no noticeable effect on engine cleanliness and wear except when the engine was firing. Under the latter conditions, fouling and wear were about quadrupled by the introduction of a small quantity of sulfur dioxide into the intake air.

When sulfur trioxide was introduced into the intake air under motoring conditions in quantities equivalent to about one-third of the sulfur introduced as sulfur dioxide, fouling was increased about tenfold and ring wear fortyfold over that obtained on firing operation on the reference fuel. Bear in mind that no fuel was being injected during the motoring tests with sulfur trioxide being introduced with the air, yet the resultant engine condition was extremely bad. The condition of the engine following the test on the sulfur trioxide is quite evident from the photographs shown in Figs. 10 and 11. The whitish material shown on the piston rings in Fig. 10 consists of metallic salts formed when water from the cooling system came into contact with the top of the piston while the engine was being disassembled. The entire crankcase was coated with the heavy sludge shown on the piston underside in Fig. 11.

On the basis of these results, it has been concluded that sulfur trioxide is the chief product of combustion of sulfur in diesel fuels and that it is responsible for the fouling and wear due to sulfur. It attacks the lubricating oil on the cylinder walls and in the crankcase forming resinous materials, which, at the high temperatures incident to engine operation, harden to form varnish and coke. The mechanism of ring and cylinder wear cannot be as definitely established from these tests. However, it appears that such wear may be due to one or more of the following:

(a) Acidic corrosion.

(b) Abrasion by hard carbon.

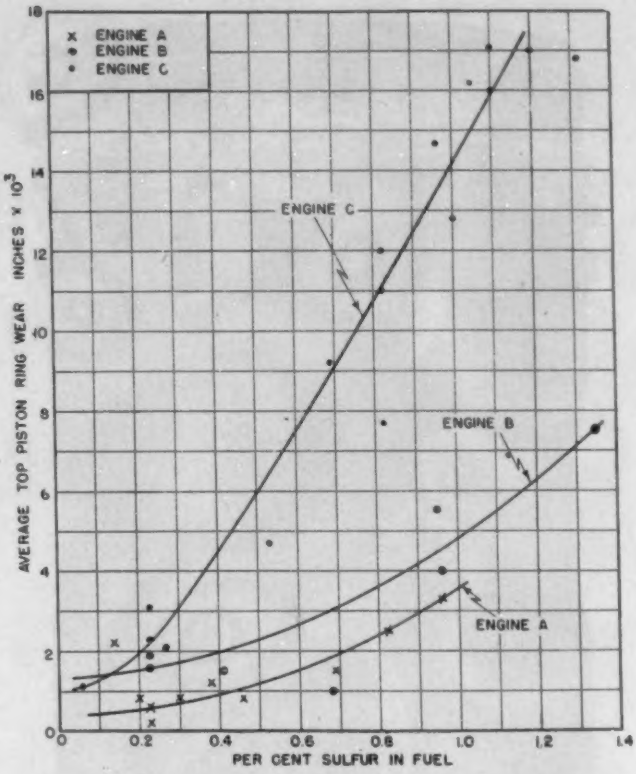
(c) Increased metallic friction caused by carbonaceous deposition in the ring grooves.

▪ Effect of Ignition Quality

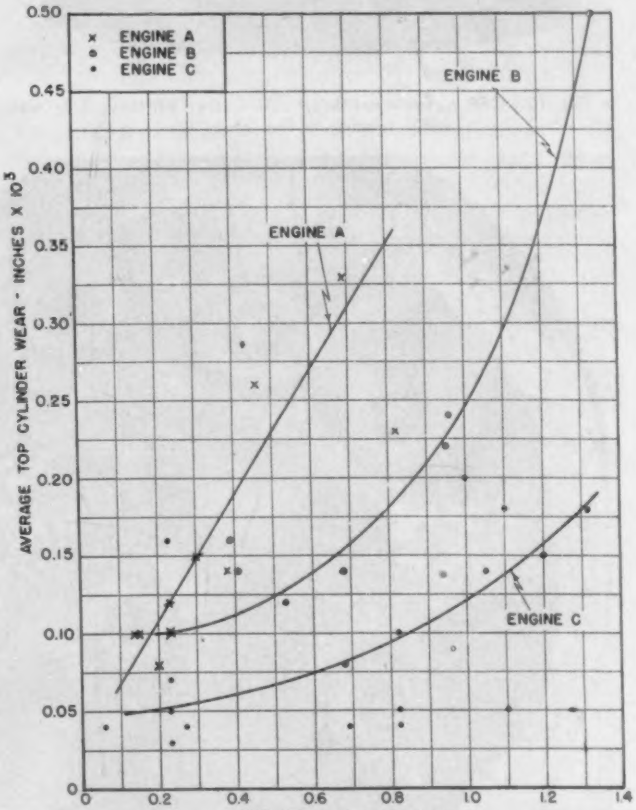
Ignition quality appears to have an appreciable effect on the performance of engines B and C and only a small effect on engine A. This is shown in Fig. 12, which gives the relation between cetane number and overall engine fouling. All three engines show an increase in fouling with decreasing cetane number. Cetane increase in the range above 45 cetane number has little effect on overall engine fouling. Below about 45 cetane number, fouling increases rather abruptly for engines B and C. General performance tests have shown that smoke and roughness become objectionable on engines A and B below about 45 cetane number. In none of the three engines did change in cetane number exhibit any discernible influence on engine wear.

▪ Effect of Viscosity and Volatility

The results of the study to determine the effect of fuel viscosity on fouling and wear have shown wide variations



▪ Fig. 8 – Effect of sulfur content of fuel on average top-piston-ring wear – 80-hr tests



▪ Fig. 9 – Effect of sulfur content of fuel on average top-cylinder wear – 80-hr tests

in the sensitivity of the various engine types to changes in viscosity. The relation between overall engine fouling

and viscosity (SUS) at 100 F is shown in Fig. 13 for engines A and C. The points represent data obtained in engine C. The dotted line showing the results on engine A was taken from a cross-plot. As is shown, the sensitivities of engines A and C to changes in fuel viscosity are substantially equivalent. A change from 35 to 40 viscosity at 100 F, which covers the range most generally specified for high-speed diesel fuels, resulted in an increase in fouling of about 27% on both engines. In the case of engine B, a three-fuel comparison was made and a change in viscosity from 35 to 66 showed no effect in overall fouling. Although the change of 35 to 40 viscosity approximately doubled ring wear in engine A, it had no perceptible effect on ring wear in engines B and C. This change of 35 to 40 viscosity did not affect cylinder wear appreciably in any of the three engines.

Inasmuch as overall volatility and viscosity are closely related for normal fuels of similar ignition quality, the relations between 50% point, 90% point, FBP, and engine fouling are similar in character to the data shown for viscosity and, therefore, have not been treated as separate factors in this paper. The presence of small amounts of "heavy" ends in diesel fuels apparently do not appreciably affect engine condition as long as they are clean stocks. In a test in engine A, the addition of sufficient distillate lubricating oil to a typical diesel fuel to raise its end-point by 100 F caused no perceptible change in engine condition. In a test in engine C, a blend of 70% kerosene and 30% light lubricating oil showed a somewhat greater fouling tendency than a normal fuel of similar viscosity and cetane number, but the difference between the two fuels was not marked. These results are in agreement with tests that showed the addition of lubricating oil to a kerosene to affect its smoking tendency only as it affected its viscosity.[4]

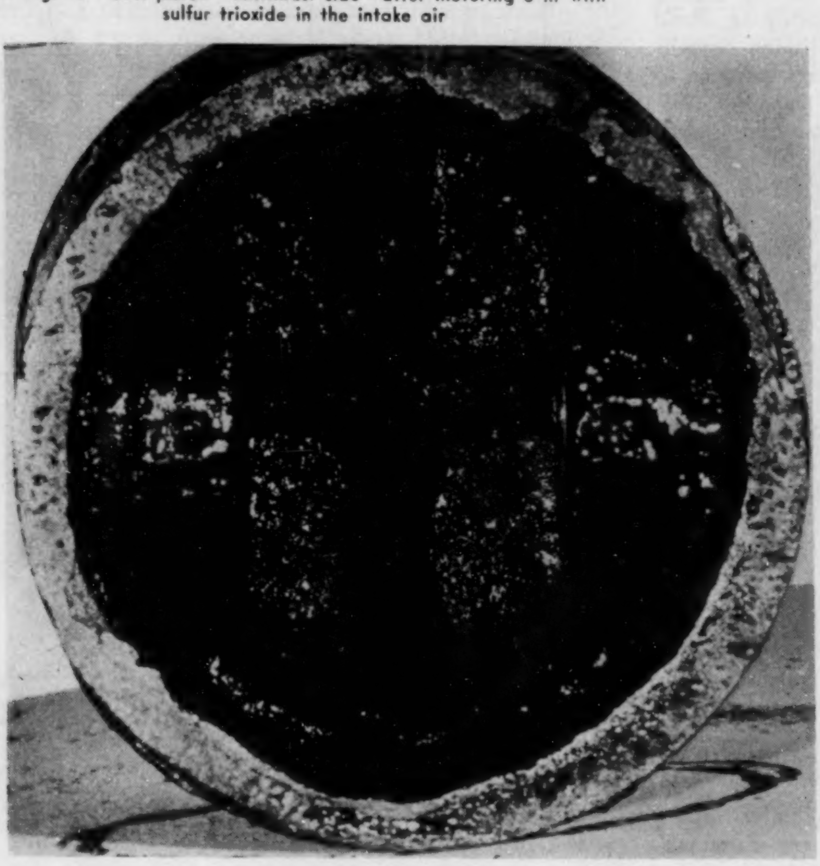
■ Fig. 10 – CFR piston – antithrust side – after motoring 8 hr with sulfur trioxide in the intake air

■ Effect of Lubricating Oil

As mentioned earlier, the lubricants used in the crankcases of engines A and C were straight mineral oils and that of engine B was a compounded oil of the detergent type. No tests were run on straight mineral oils in engine B for comparison with the results shown for the compounded oil. However, comparative tests on the two types of lubricants were run in engine C. As shown in Fig. 14, the addition of a detergent-type compounded to the lubricant markedly reduced fouling in engine C on both the high and low sulfur fuels and to a great extent washed out the differences between the fuels due to sulfur. Although the use of the lubricant additive also reduced ring and cylinder wear to about half that obtained on the straight mineral oil, changing from the low to the high sulfur fuel still increased ring wear four-

[4] See SAE Transactions, Vol. 47, October, 1940, pp. 397-406: "The Control of Smoke in the Automotive Diesel," by W. W. Manville, G. H. Cloud, A. J. Blackwood, and W. J. Sweeney.

turn to page 419



■ Fig. 11 – CFR piston – underside – after motoring 8 hr with sulfur trioxide in the intake air

IMPACT EXTRUSION and COLD PRESSING of Airplane Parts

by PHIL KOENIG
Consolidated Vultee Aircraft Corp.

FOR many years certain articles, such as collapsible tubes and similar containers as used for holding semiliquid materials – tooth paste, shaving creams, and paints, for example, have been made of tin, lead, and other soft metals. The method used has been the impact-extrusion process. Any other method would be highly undesirable, for it would be far more elaborate, much slower, and extremely expensive in comparison.

With the impact-extrusion method, a single press stroke changes a slug or blank of metal into the completed article. The high pressure developed causes the cold metal to flow to the desired shape.

Technical progress in the field of creative science and in the science of production should be the aim of everyone connected with war production. Our machines must not only be used to the best manufacturing advantage, but expert consideration must be given to the substitution of methods which are necessitated as a result of limited production in other lines and on account of scarcity of materials.

Many airplane parts which would normally be produced by casting, forging, or by machining from solid stock, can be produced very rapidly and extremely economically by the impact-extrusion method.

[This paper was presented at the SAE National Aircraft Production Meeting, Los Angeles, Calif., Oct. 1, 1942.]

Aluminum and aluminum alloys can be extruded by the impact method. The only limit to the size of the parts so produced is the power of the press that can be obtained for the work. Small parts can be produced on standard crank presses which are available in almost every shop. Any type of press can be used, as it is not necessary to have a high speed, except, of course, in respect to production capacity. Hydraulic presses can be used to advantage. Small parts can be produced in multiple dies and, in this way, much larger production obtained. It has been found by experiment that aluminum requires a pressure of approximately 45 tons per sq in. for this production method while 24S aluminum alloy requires a pressure of approximately 90 tons per sq in.

Impact extrusion can be used advantageously as a method of producing a very satisfactory substitute for many parts that are now being made by the drop-hammer process. Because of the increasing difficulty in obtaining forgings, this substitute method should be rapidly promoted and used to the maximum extent possible.

The dies for the manufacture of parts produced by the impact-extrusion method must have sufficient strength and proper heat-treatment to withstand the extreme pressures and impact shocks under the continuous operation imposed. During the flow period through which the metal passes, there is, as a consequence of the high pressures

DEVELOPMENTS resulting from research inspired by the war's demands for greater speed, larger volume, and material and labor conservation, have led to the manufacture of many airplane parts by the impact-extrusion method.

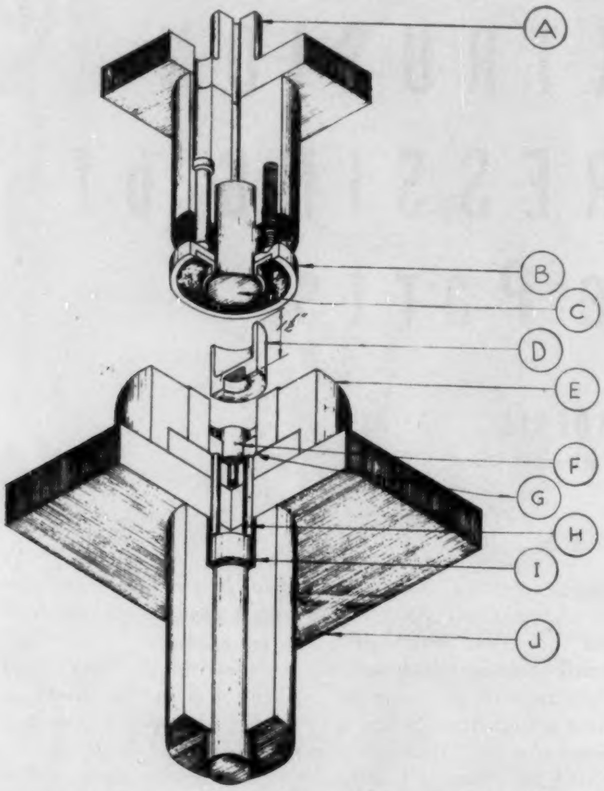
Where the previous methods called for casting, forging, or machining from solid stock, research has developed ways to use the impact-extrusion method that are more rapid and economical.

Aluminum and aluminum alloys can be extruded by this method and their size is limited only by the power of the press available for the work. Small parts are produced in large quantities by the use of multiple dies.

Experiments have established the pressures required to form these materials by the impact-extrusion method, complicated designs and shapes can be easily produced, and there seems to be no limit to the height to which the metal will flow, if the required force is applied to the tools.

★ ★ ★

THE AUTHOR: PHIL KOENIG, supervisor of tooling at Consolidated Vultee Aircraft's New Orleans Division, speaks from a rich background of experience. He has been master mechanic, machine shop foreman, tool designer and chief tool engineer. He is a native of Buffalo.



A Punch assembly *B* Stripper plate *C* Punch insert
D Part *E* Die assembly *F* Die form punch
G Stripper pad *H* Stripper pins *I* Stripper shaft
J Die shoe

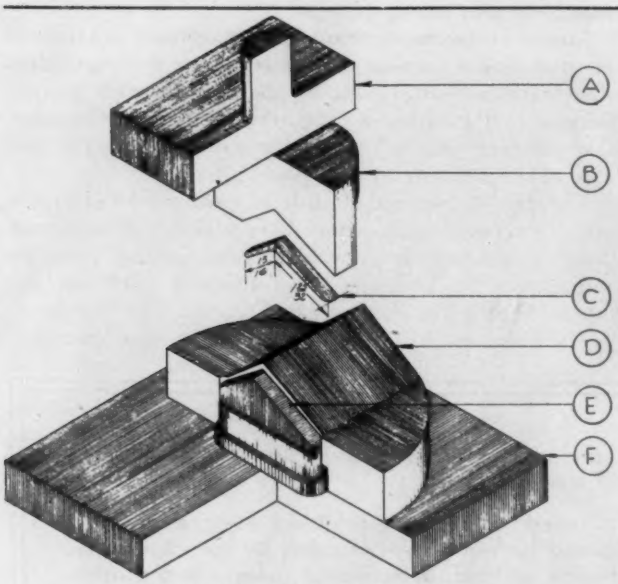
■ **Figs. 1 (left) and 2 (above) – An impact-extrusion die and its formed part**

This particular part is a roller on the ammunition feed. There are quite a number of these parts required per ship. The material used for fabrication of this part is 1½" diameter aluminum bar in the soft condition. The stock is cut to the required developed length on the automatic screw machine. These blanks are then placed in the die and, with a single stroke of the press, converted into a completed part with the exception of drilling a ¾" hole and facing the end on a turret lathe. Approximately 55% less material is required for the impact-extrusion method as compared to turning the part on a turret lathe. A time study reveals the time saved as follows:

Part made complete on turret lathe: 25 sec.

Part made by impact extrusion and finished on turret lathe: 20 sec.

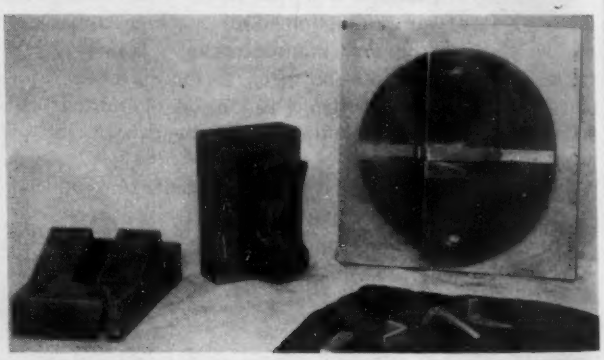
The 0.9024 $\begin{matrix}+0.001\\-0.000\end{matrix}$ dimension on the bore is held much closer by the extrusion method than would be possible on the turret lathe.



A Punch holder *B* Punch *C* Part
D Die *E* Die insert *F* Die plate

■ **Figs. 3 (above) and 4 (right) – A die that can be called either an impact-extrusion or a coining die**

This die was the first one to be made; considerable difficulty was experienced in the development of it, as the first die was not strong enough and there was too much clearance between the die knock-out and the opening in the die, which created excessive flash.



Alclad was tried with the result that most of the aluminum was forced in the radius of the inner corner which, of course, weakened the part considerably.

Flat stock was first tried, and in comfining the metal in order to obtain semisharp outer corners, cold shuts in the inner radius resulted.

Consideration should be given, in cases of this character, to this distribution of the metal, as in the case of drop-forging practice. There, a first die, or "fuller," is sometimes necessary to create metal distribution that will ensure filling the roughing and finishing impressions of the dies.

In the case of this impact-extruded part, the slug should have a cross-section at least equal to the thickness of the finished part. That is, the dimension from corner to fillet.

In the case in question, when a slug of round stock having ample diameter was used, metal flowed in both directions and perfect parts resulted.

between the die surfaces and the flowing metal, considerable friction and, as a consequence, wear to be considered. It has been found that a plating of chromium is advantageous, as it materially lengthens the life of the dies.

There are many advantages to this form of production. These can be roughly enumerated as follows:

1. The density of the metal is considerably increased; consequently, the metal is not harmed by this process. On the other hand, a material improvement is brought about.

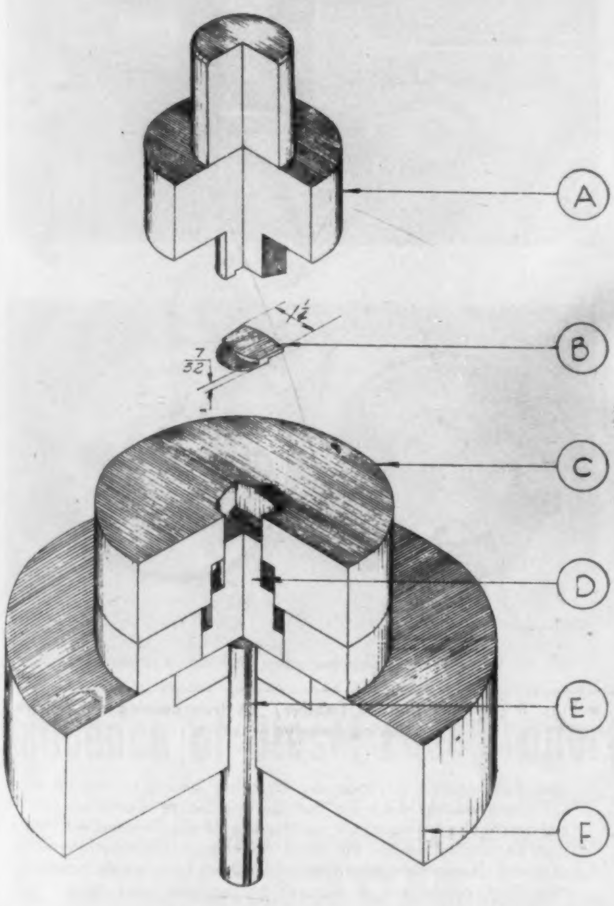
2. The tool cost is relatively low. The dies, in the majority of cases, can be easily machined. There is no special skill required in the production of these dies over those of other methods. Tools for this method do not require any special care in the course of their creation. There are only a comparatively few and simple "rules" to be followed.

3. Very low production cost is obtained, as skilled labor is unnecessary. Parts produced by impact extrusion, once the dies are created, require merely the insertion of the blank or slug in the die cavity and the removal of the finished parts after the power stroke has been completed. Both of these could be done by means of a properly designed feeding mechanism. A combination of this character would, although only needed for large quantity, be extremely efficient.

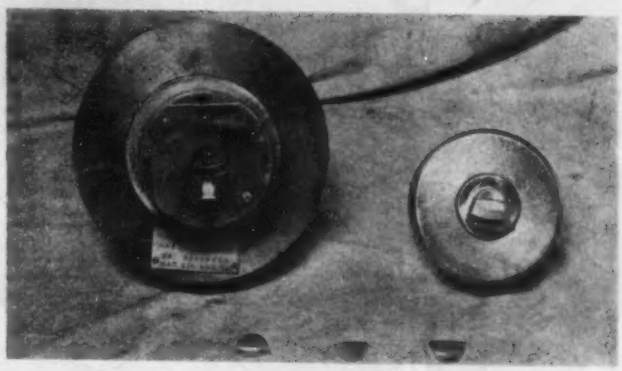
4. Extremely close tolerances are automatically obtained. Tolerances are, when using this method, of no special consideration. This is radically different, in this respect, from any other system. Here, close tolerances are ensured and at no expenditure of time, either in the making of the parts or the inspection of the product. If the tools are made correctly, the parts will be within the required dimensions.

5. There are no set limitations in regard to the form or shape of the parts which can be produced by the impact method. Symmetrical and cylindrical shapes might be best suited for production as far as the tools are concerned. Square, rectangular, and oval, as well as unsymmetrical shapes, can be produced with equal ease after the tools have been made. There is no limit to the height to which a slug can be driven. The stroke of the press might control this but the metal will flow to any extreme if the required force is applied to the tools.

There is another consideration that might be given here,



A Punch *B* Part *C* Die
D Die insert *E* Stripper shaft *F* Die shoe



This process requires three operations: blanking, coining, and drilling; compared to the old process, which consisted of blanking, two separate turning operations, and drilling.

A considerable amount of time can be saved, while at the same time more accurate parts are obtained, requiring only inexperienced help.

This is an example of a true coining die. There is no overflow of metal, which consequently necessitates that blanks be of proper thickness and hardness, as any deviation will result in a broken die.

■ Figs. 5 (above), 6 and 7 (right) – A coin-press die for fabricating parts of 24S dural

namely, that of producing parts with holes or openings. Sometimes it might be best to drill, punch, or otherwise produce these after the part has been otherwise completed but, when this is not the case, the hole or opening should be made in the slug or blank to correspond and, where proper allowances are made, success is ensured.

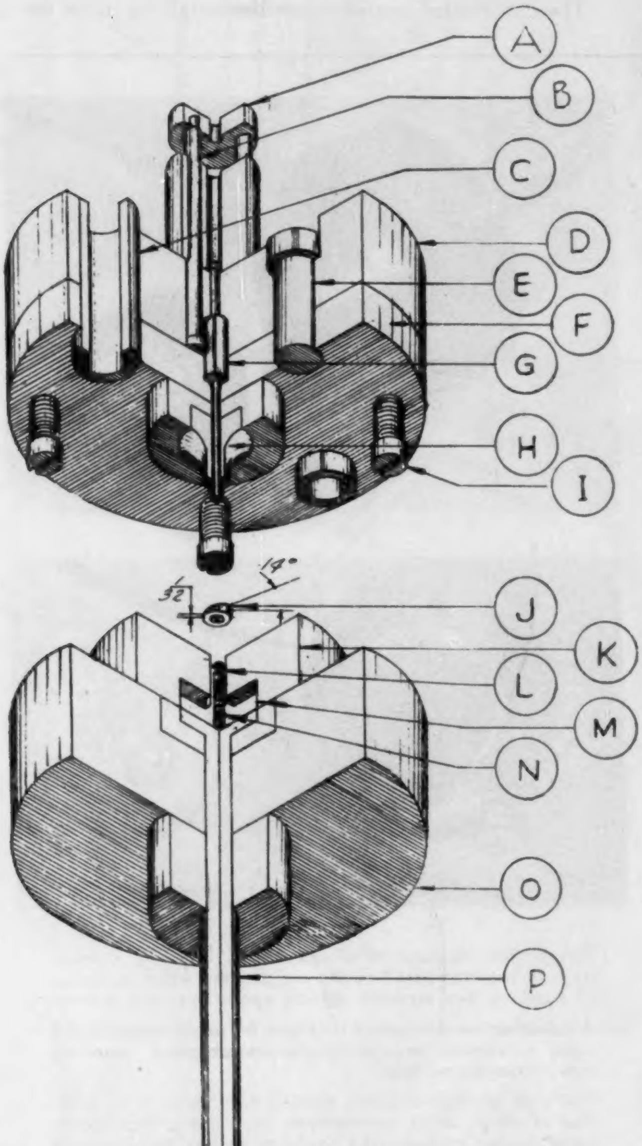
Care must be given, it has been found, to the design and fit of all dies, so as to prevent the metal being formed from being forced into the clearance spaces between the die body and the knock-out plunger. The high pressures which are involved necessitate close fits in this respect, but no special difficulty need be expected here.

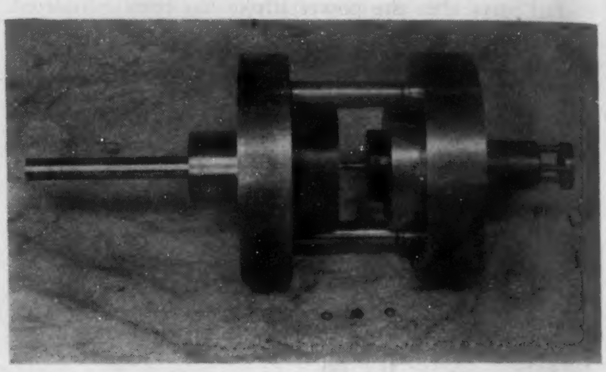
6. Secondary operations, wherever necessary, can greatly broaden the field to be covered by this system of production. In summary, it has been prophesied, we believe correctly, that impact extruding will be used extensively in the production of aeronautical parts. This will not only aid materially in obtaining items otherwise requiring production by the drop-hammer method, but will be the means of more efficiently producing parts made on the turret lathe, milling machine, or other similar methods.

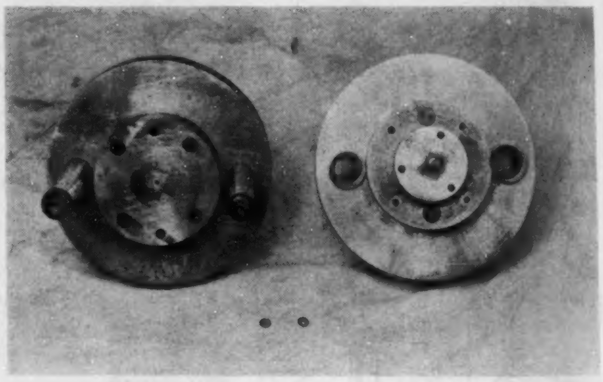
7. Thermal methods can be used to supplement the strain-hardening improvement brought about by this method.

8. Complicated designs and shapes can be produced which would be very difficult, or impossible, to produce by any other method and, with properly constructed dies, they are obtained with the same perfect ease as the simplest pieces.

In these hectic days wherein skilled help is at a premium, wartime emergency demands the use of every facility and every method that Yankee ingenuity and industrial development can bring about. This is one of a number that have already been put into production and that will help in the dire competition which exists between our scientific and productive efforts and those of our enemies.





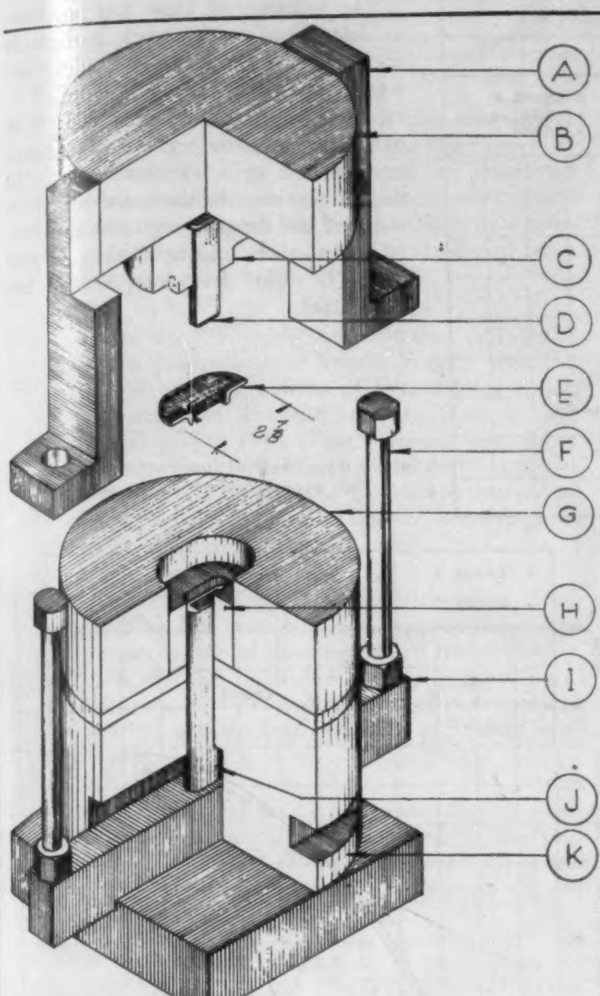


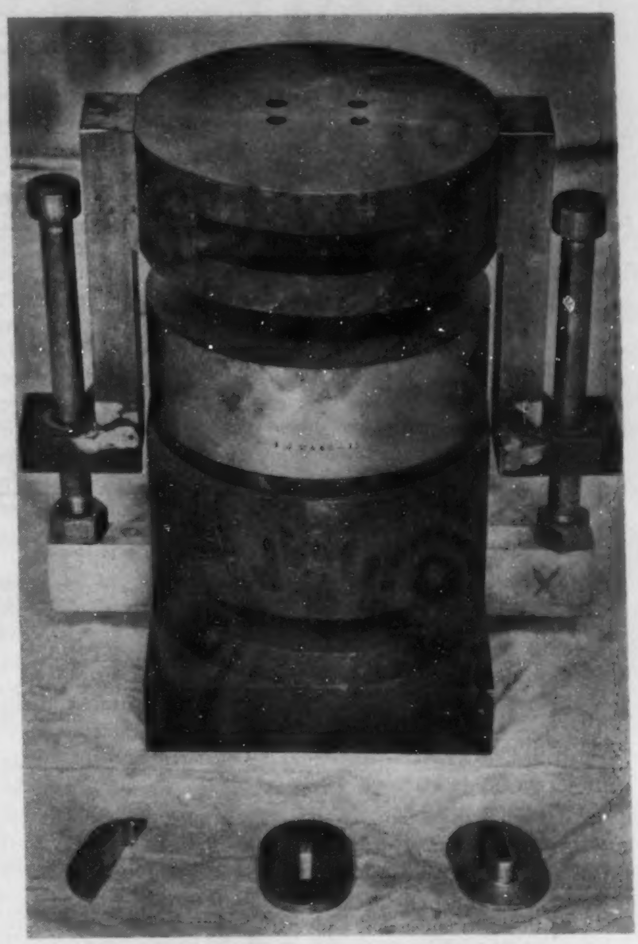
■ Figs. 8 (left), 9 and 10 (above) – A true coining die for producing beveled washers.

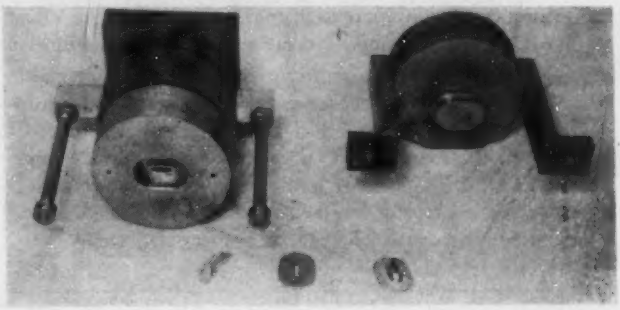
The flat washer is made by blanking and piercing on a standard washer die. The outside and inside diameters are not particularly important, as the size of the finished washer can be controlled by the stock thickness. The coining die converts these blanks into bevel washers in a single operation that produces an exceedingly uniform and accurate part.

■

A Punch plate *B* Punch pin *C* Guide bushings
D Punch shoe *E* Guide pin *F* Punch holder
G Locating pin *H* Punch insert *I* Stripper spring screws
J Part *K* Die holder *L* Locating pin
M Die *N* Locating pin spring *O* Die shoe
P Knock-out shaft







A Stripper bar *B* Punch shoe *C* Punch holder
D Punch *E* Part *F* Stripper bolt
G Die block *H* Die *I* Stripper bar
J Knock-out *K* Die shoe

■

▪ Figs. 11 (above), 12 and 13 (right) – An impact-extrusion die for production of a part of 24S dural that would otherwise be a drop forging

With this method of production, the important dimensions are held to very close tolerances at no increase in labor cost. A part of this character would require no machining except, perhaps, for any required holes and machining the end of the elongated section, as this is the area which is influenced by any excess metal which might be in the slug.

Influence of Diesel Fuel Properties on Engine Deposits and Wear

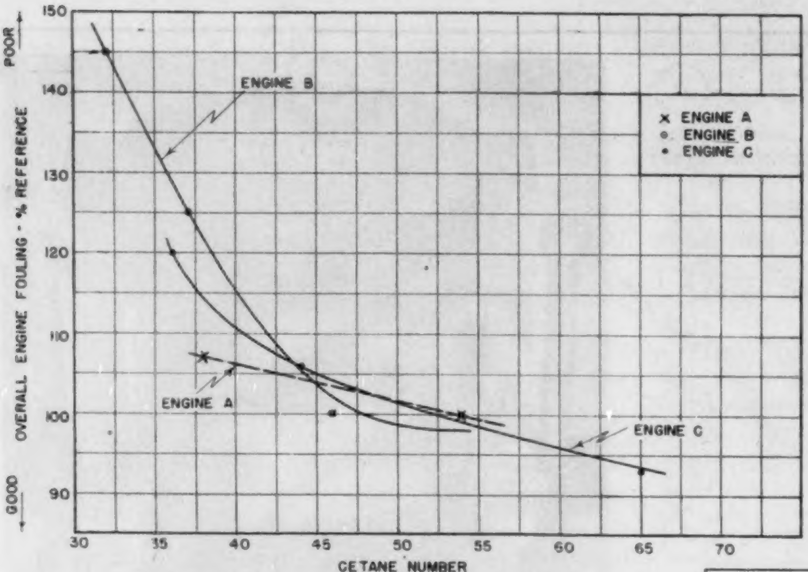
continued from page 414

fold and doubled cylinder wear. Field tests on compounded lubricants of the detergent type have verified those data. The lubricants used are briefly identified in Table 3.

Considering all phases of these studies, it is clear that the selection of suitable fuels is an important factor in the control of fouling and wear of automotive diesel engines. Assuming stability, freedom from water and soaps, and absence of corrosive, abrasive, and residual materials, the major diesel fuel characteristics governing automotive diesel-engine fouling and wear are sulfur content, ignition quality, viscosity, and/or volatility.

Table 3 – Lubricants Used in Fouling Studies

Inspections	Naphthenic Stock "A" (Acid Treated)	Naphthenic "A" + Additive	Naphthenic Stock "B" + Additive	Paraffinic Stock
Gravity, API	23.0	22.5	21.2	29.2
Flash, F	405	405	390	425
Viscosity at 210 F, SUS	55.8	56.6	57.5	53.6
Viscosity Index	42	44	10	101
Color (R)	14½	9¼	2	9¼
Conradson Carbon	0.03	0.42	0.31	0.10



■ Fig. 12 – Effect of ignition quality on overall engine fouling – 80-hr tests

Of these properties, sulfur content is the most important. Sulfur in diesel fuels is converted during combustion in the engine, chiefly to sulfur trioxide, which attacks the lubricating oil producing insoluble sludge that results in varnish and carbon formation and in increased ring and cylinder wear.

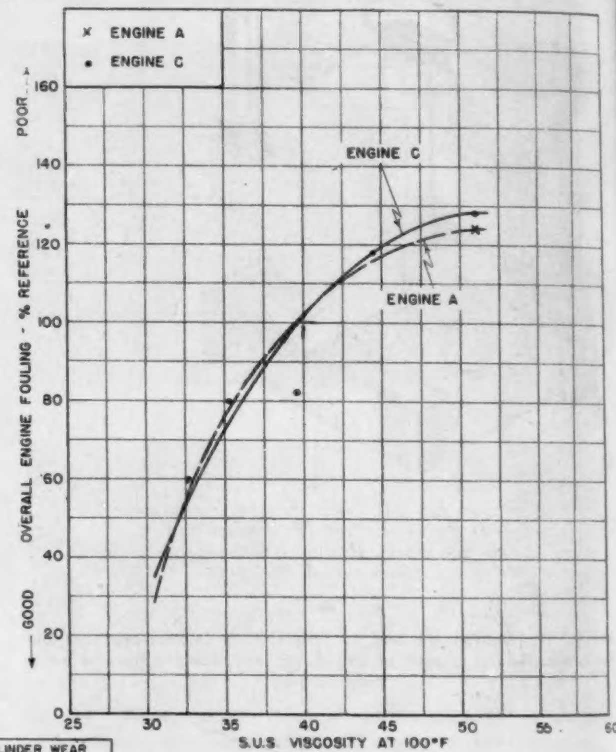
As regards ignition quality, the minimum cetane number acceptable will depend to a great extent on the sensitivity of the engine under consideration. The data indicate that little improvement in engine condition results when the cetane number is increased above that necessary for smooth operation. Reducing cetane number below this satisfactory minimum results in increased engine fouling. The effect of cetane number on engine wear appears to be negligible.

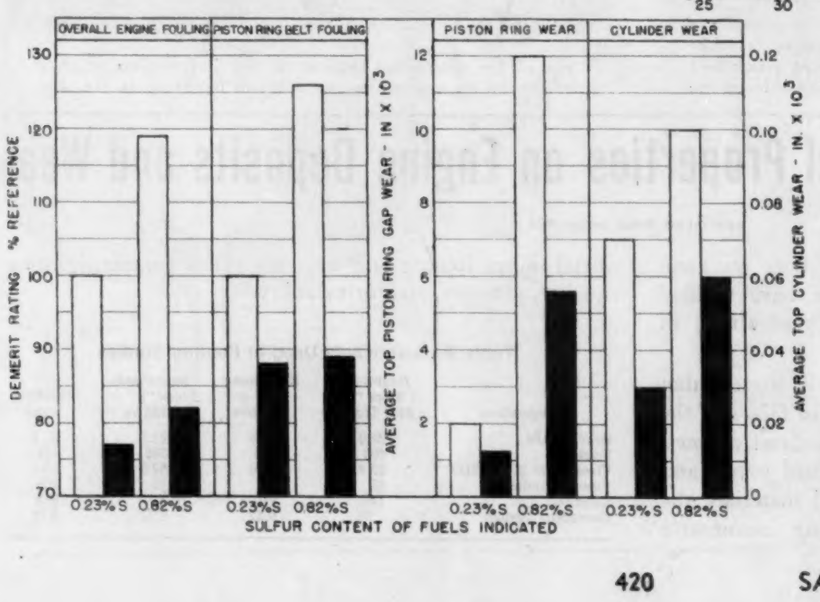
Fuel viscosity and volatility may show an appreciable effect on fouling and wear in certain types of diesel equipment. The very close relation between these two factors in normal fuels makes it unnecessary to specify both rigidly. The limitations generally put on these characteristics for the control of smoke, and so forth, appear to be sufficient for the control of engine fouling and wear.

The influence of these fuel properties as determined in this work, particularly the influence of sulfur on fouling and wear, apparently cannot be applied directly to larger slow-speed diesels in marine or industrial service. A 12-month test on two 13 x 18 diesel engines in marine service – the starboard engine on one fuel and the port engine on the other – failed to show unusual fouling or wear on high sulfur fuel compared to low sulfur fuel.

■

■ Fig. 13 – Effect of fuel viscosity on overall engine fouling – 80-hr tests





■ Acknowledgment

The authors wish to acknowledge the very great assistance of A. D. Haff of the Bethlehem Shipbuilding Co., formerly with the Esso Laboratories, Research Division, in the preparation of this paper.

■

■ Fig. 14 – Effect of lubricant on engine fouling and wear – engine C, 80-hr tests (white blocks represent uncompounded lubricant, black blocks represent compounded lubricant)

TODAY'S inspection procedures are much more rigorous, as well as more rapid, than those of 25 years ago. The development of improved standards, refinements in old methods of inspecting parts, and the application of entirely new principles and tests to the science of precision have all contributed to this progress.

To contrast the highly scientific and systematic inspection procedures of World War II with the crude and slow inspections of World War I, Mr. Snider compares the methods applied to the crankshaft of the 1918 Liberty engine with those applied to the 1943 Allison crankshaft – an aircraft crankshaft being used because it covers almost all types of measurements employed in checking machined parts.

During the period under consideration, steel composition has not changed appreciably, and yet, results obtained have improved tremendously. This progress has been due largely to more careful selection of raw materials by the transverse fatigue test and the improvements in finished heat treatment.

Poorly controlled only a few years ago, today's heat-treatment results have been tied to finer and more definite units of measurement. Checking case depth and surface hardness through the use of better machines has contributed to uniformity and durability of the final product.

Grain size is one of the items that have become more important than in 1918, fine-grain steels now being specified because of better life characteristics.

Inspections start with the forging billets and continue at frequent intervals in the fabrication of the crankshaft to the final visual and dimensional check of the finished part. Checking is done at the machine where possible, but other measurements impractical to check there are made at inspection stations. Precision devices formerly available only to the toolroom or laboratory are now being used regularly at these stations.

New developments in precision instruments that are used in checking the crankshaft include master steel block gages, magnetic inspection, and electric gages. Several other valuable developments are discussed by Mr. Snider, although they are not used in crankshaft inspection.

The production of interchangeable parts in commercial quantities has also been possible because of the improvements in every branch of machine tool engineering. As the means of measurement has improved, so has the degree of accuracy of production increased.

■ ■ ■

THE AUTHOR: O. J. SNIDER, who has been with Cadillac Motor Car Division since 1932, is now general superintendent of inspection. Having started his career in the automotive industry as an assembler with Fairbanks, Morse & Co. at the age of 17, Mr. Snider rose to his present position by working in the field as draftsman, pattern maker, superintendent of production, designer, chief tool inspector, and general foreman of the tool room.

PROGRESS in PRECISION

by O. J. SNIDER

Cadillac Motor Car Division, General Motors Corp.

PROGRESS in precision has been marked by the development of improved standards and better means of comparison of an article with those standards. To indicate the progress made recently, we have selected as a means of comparison a specific part which will be used to show and describe some of the standards and some of the methods of measurement against those standards. The time element will be that between World War I up to the present.

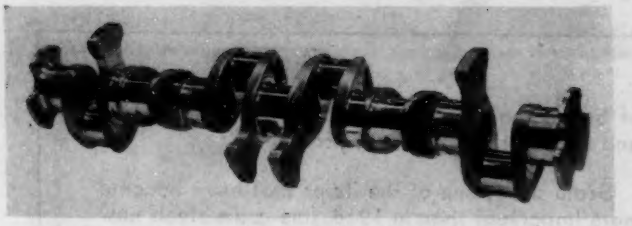
The part selected is an aircraft crankshaft (Fig. 1), which covers practically all types of measurements used in checking a part that is machined. The various steps will be presented in chronological order, and where possible previous methods will be indicated and comparisons drawn with present-day methods.

The crankshaft is the most highly stressed major part in an engine. For this reason, physical-property specifications, dependent on the steel mill and forge shop, size, balance, finish, and hardness must be controlled to a standard in order to ensure efficient engine operation.

Crankshafts designed for aircraft engines must be able to transmit more power for a given weight than crankshafts designed for other engines. Vibrational, torsional, and bending stresses of great magnitude are developed, yet the

---

[This paper was presented at the SAE War Materiel Meeting, Detroit, Mich., June 9, 1943.]



■ Fig. 1 – The aircraft crankshaft covers practically all types of measurements used in checking a machined part

weight-per-horsepower ratio must be held to a minimum, without sacrificing strength.

Failure to maintain quality standards on the crankshaft might be the direct cause of a functional engine failure, and may result in loss of the plane and the pilot. Of still more vital importance, major battles may be won or lost, depending upon the performance of one engine. It might even be a direct cause in the final outcome of a battle and the war.

There are various functions affecting the quality of an aircraft-engine crankshaft. The foremost consideration in actual engine operation is durability, which is influenced by the metallurgical properties of the forging, as well as the dimensional control of the product.

Let us look at the requirements which must be checked on the Allison engine today in comparison with the requirements of the Liberty engine of 1918. These data are shown in Table 1.

Table 1 – Comparative Data – Aircraft Engines

	Liberty V12, 1918	Allison V12, 1943
Hp (take-off)	400 at 1700 rpm	1425 at 3000 rpm
Hp at 10,000 ft	275	Somewhat below above figure. Depends upon supercharger used
Hp at 25,000 ft	100	
Weight per Hp, lb	2.3	0.93
Displacement, cu in.	1650	1710
Reduction Gear Ratio	None	Approximately 2:1
Bore and Stroke, in.	5 x 7	5½ x 6
Time between Overhauls, hr	50	Approximately 10 times the Liberty time

Note that the horsepower is increased approximately 360%. The weight per horsepower decreased 60%. The period between overhauls and the reliability of the engine have increased many times.

Table 2 gives crankshaft data which indicate other differences in these two products. Note that the crankshaft weight has decreased 10% while the power has increased

Table 2 – Metallurgical Comparisons of Aircraft Crankshafts

	Liberty, V12, 1918	Allison V12, 1943
Crankshaft Weight, lb	105	95
Weight per Hp, lb	0.26	0.065
Crankshaft Steels	X3340, 3240, 6140, 3140	AMS 6415 (X4340)
Heat Treatment	Quench, temper, machine	Quench, temper, machine, stress relieve, nitride
Physical Properties:		
Minimum Tensile Strength, psi	135,000	140,000
Minimum Yield Strength, psi	120,000	120,000
Minimum Elongation, %	16	15
Minimum Reduction in Area, %	50	50
Izod Impact, ft-lb	40	35
Brinell	284-331 (Estimated)	302-341
Case Hardness	....	Vickers 500 minimum
Case Depth	None	0.020 minimum
Grain Size	Not specified	5 – 8
Cleanliness	Not specified	Magnetic inspection quality

to 360%. The crank weight per horsepower has decreased 75%.

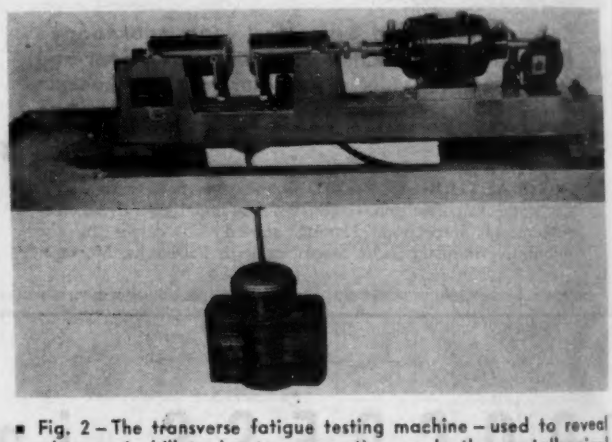
Assurance that each crankshaft meets the standards requires inspection from the very start.

Inspection procedure today likewise is more vigorous, as indicated by Table 3, which gives a comparison of forging billet inspection. The 1943 inspection procedures include several new methods and tests.

Table 3 – Forging Billet Inspection

	Liberty. V12, 1918	Allison V12, 1943
Segregation or Soundness	Deep etch on cross-sections of billets	Deep etch on cross-sections of billets. Transverse fatigue test (six specimens from each heat)
Chemical Composition	Chemical analysis	Chemical analysis Spectrographic analysis
Grain Size	No standards or procedures for test	McQuaid Ehn test for testing grain size
Hardenability	No standards or procedures for test	End-quench method for measuring hardenability; also other methods

The transverse fatigue test is applied to specimens cut from a billet section which has been heat-treated to a specified Brinell hardness. This test will reveal weaknesses in billets due to segregations and other metallurgical defects. (Fig. 2.)



■ Fig. 2 – The transverse fatigue testing machine – used to reveal weaknesses in billets due to segregations and other metallurgical defects

Tests are made from billets representing top and bottom of several ingots in the heat. (Fig. 3.)

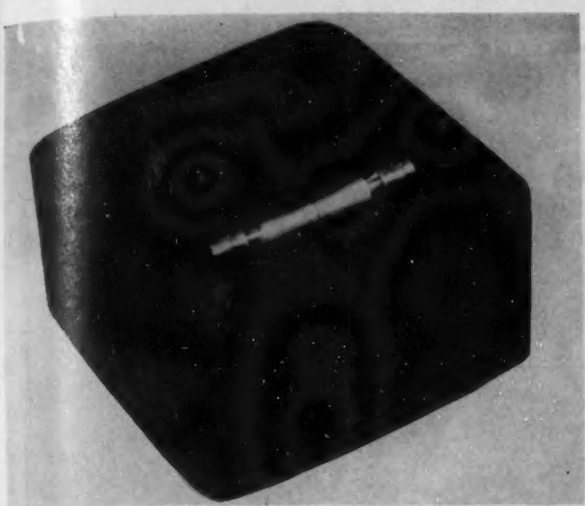
Grain-size control has grown up between World War I and II. Fine-grained steels are now specified because of their improved life characteristics, and are checked by the steel mill, forge plant, and the purchaser.

The hardening characteristics of each particular heat of steel are checked by the end-quench hardenability test, which has been developed in the past 10 years.

Following the billet inspection the forgings are made, using carefully controlled procedures. The finished forgings are then inspected again. (Table 4.)

The flow lines are indicated on Fig. 4. The crankshaft is made of 4340 steel, but the specimen for checking flow lines is made from 1045 steel, which has greater segregations and etches up sharper to disclose the flow better.

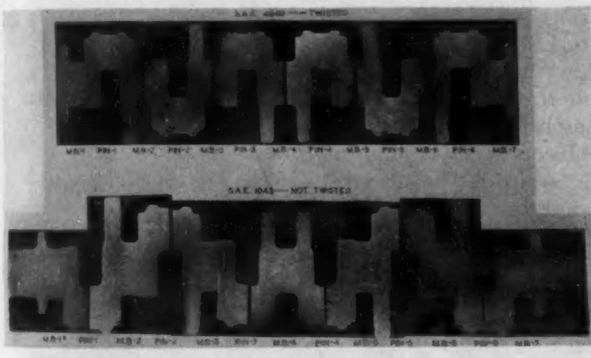
The forgings are normalized, hardened, and tempered, and the test coupon cut from one end. Tests made include



■ Fig. 3 – Test specimens are taken from sections of billets representing the top and the bottom of several ingots in the heat

Table 4 – Forging Inspection

	Liberty V12, 1918	Allison V12, 1943
Forging Defects and Soundness	Visual inspection	Visual plus magnetic inspection
Flow-Line Examination	No inspection	Deep etch to check flow lines. Check on forging practice occasionally



■ Fig. 4 – Grain flow in Allison crankshaft – the part is made of SAE 4340 (above), but the specimen for checking flow lines is made from SAE 1045 (below), which has greater segregations and etches up sharper to disclose the flow better

tensile, Izod, fracture appearance, and Brinell hardness of crank and coupon.

Each forging and coupon is stamped with an identification number. The forge shop prepares affidavits listing all test data.

It will have been noted up to this point in following our specific part, that the steel composition has not changed appreciably over this period, and yet the results obtained have been improved tremendously. This has been due largely to progress in careful selection of raw material by the transverse fatigue test and the improvement in finished heat treatment.

Adherence to the standards established for the precision part is required during the processing stage. This is accomplished by incorporating inspection procedure in the routing of the part, so that inspection is made adjacent to the machines where the operations are performed. In this manner the production department is assisted in manufacturing a precision part.

In recent years the differences in floor inspection shown in Table 5 have developed.

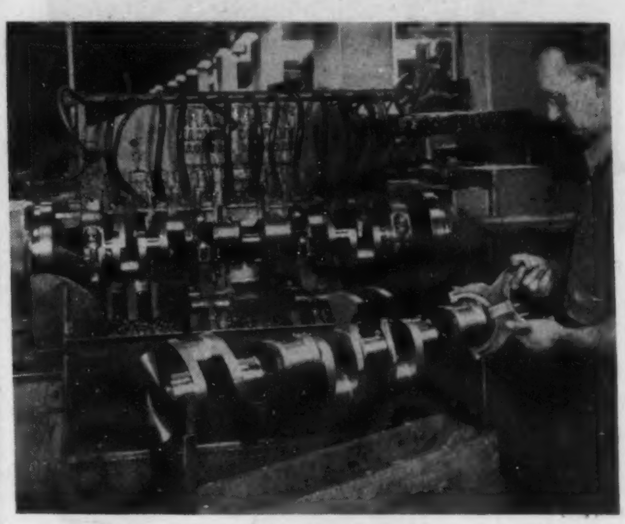
Table 5 – Comparison of Floor Inspections

Then	Now
Highly skilled help	Specialized help (including women)
Micrometers and calipers	Electric gages
Paddle gages	Paddle gages and air gages
Height gages	Fixed step gages
Verniers	Special gages, comparators – electric and mechanical
Double-end plug, snap, and bar gages	Progressive gages (double-end interchangeable gages)
Selective assembly	Greater interchangeability
Greater skill with more time and less accuracy	More efficiency and accuracy with modern optical and electrical instruments
	Longer life gages, diamonds, and carbide tipped instruments

In checking various rough and semifinish machining operations for dimensions, general use gages, such as snap, plug, bore, and indicator gages are being used. Fixed limit gages have minimized the use of micrometers and height gages, which has helped maintain uniform tolerances.

Operators are furnished quick, accurate gages to be used right at the machine. (Fig. 5.) Progressive gages have supplemented the use of sine bar, Jo blocks, and vernier height gages, affording more checks with greater accuracy in less time.

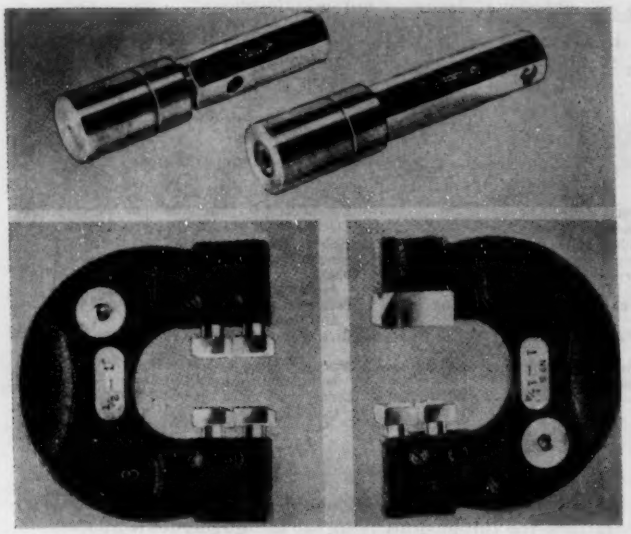
Progressive gages as shown in Fig. 6 are comparatively new. They are used widely to permit faster checking.



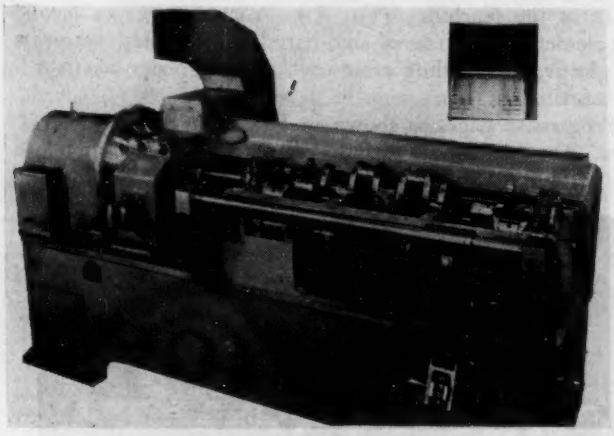
■ Fig. 5 – Operators are furnished with gages to be used right at the machine

Balancing machine principle is that of rotating a crank in a rocking cradle, the motion of which, due to errors in balance in the rotating part, is magnified and optically indicated on a screen in front of the operator, as shown in Fig. 7. Both amount and position of the off-balance mass are easily identified.

Contrast this with the old method of rolling the part on horizontal straight edges without any optical means of identifying errors. This was a crude method, and could indicate static out-of-balance only, while the present method will show both static and dynamic characteristics.



▪ Fig. 6 – Progressive gages are used widely to permit faster checking



▪ Fig. 7 – Balancing machine used to show both static and dynamic characteristics

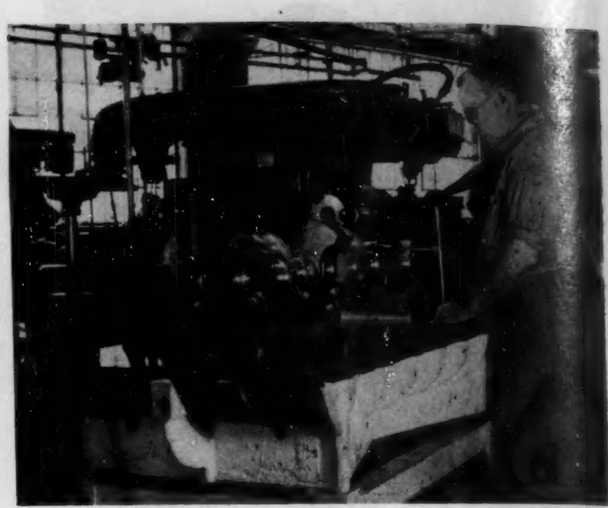
Maintaining tolerances at the machine is a step toward progress. Valuable in controlling machining limits from one operation to another, it has permitted the building of larger floats of parts so necessary to mass production. In addition, it has contributed to the saving of tools and material.

At frequent intervals, usually at the end of a certain phase of operations, the part will be checked at stations. Relationship of all dimensions processed and those measurements impractical to check at the machine are checked here. Precision devices formerly available to the toolroom or laboratory are now used regularly at such stations.

Fig. 8 shows a close-up view of a station equipped to release a crankshaft for further processing.

Station inspection, however, is not concerned exclusively with dimensional checks. A visual inspection for tool marks, improper blending, and sharp edges is needed to detect any indications which may later develop into fractures. To obtain correct finish in the final part, each processing step must be controlled. We are assured of a quality part with a minimum of repair later on.

To get a better idea of the progress in the control of



▪ Fig. 8 – At frequent intervals, the part is checked at inspection stations

heat-treatment on the crankshaft, a short description of the procedure is as follows:

The crank is purchased in the normalized, quenched, and tempered condition. Following rough machining and drilling, it is given a stress-relief tempering operation. (Four hours at 1050 F in a circulating air furnace.) It is then checked for distortion by placing the No. 1 and No. 7 bearings on rolls and checking the runout of the No. 4 bearing.

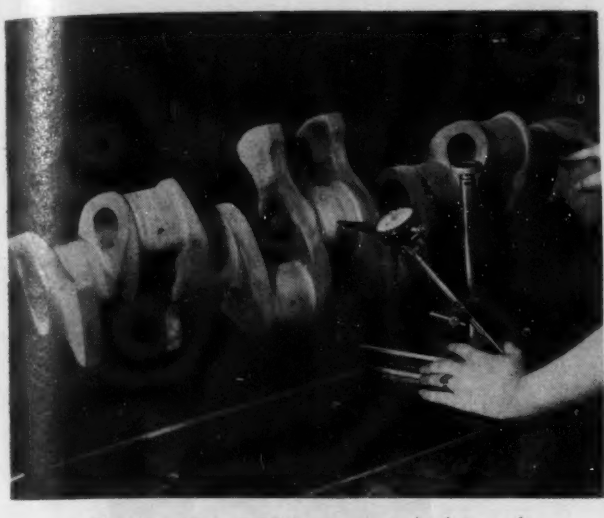
After finish machining and rough grinding, the shaft is again stress-relief tempered, this time in a nonoxidizing atmosphere.

Subsequent to semifinish grinding and polishing, the shaft is nitrided for 38 hr at 950 F. Fig. 9 shows a load of crankshafts in a nitriding furnace.

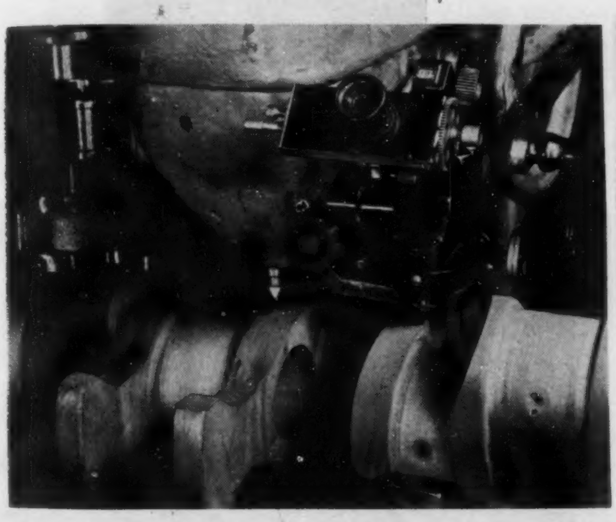
▪ Fig. 9 – Load of crankshafts in a nitriding furnace

Runout check is made after nitriding, the limit being 0.007 in. (Fig. 10.) Two stress-relieving operations given the shafts during machining leave the shafts relatively stress-free. Shafts seldom exceed the runout limits.

Also, after nitriding, checks must be made to determine if the shafts have the proper hardness and depth of hardness. (Fig. 11.) The surface hardness of each shaft is checked on a Vickers hardness tester, using a 10 kg load.



■ Fig. 10 – After nitriding, a runout check is made



■ Fig. 11 – After nitriding, the crankshaft must be checked for proper hardness and depth of hardness

Each shaft must have a Vickers hardness of 500 minimum on the surface.

The case depth is checked on a test specimen, which is nitrided with the crankshafts. This test specimen is made from the same type of steel and heat-treated to the same hardness as the crankshafts.

To prove that the proper case depth is obtained, the test specimen must have a Vickers hardness of 400 minimum after 0.020 in. is ground from the surface.

Poorly controlled only a few years ago, today's heat-treatment results have been tied to finer and more definite units of measurement. Checking case depth and surface hardness through the use of improved machines has contributed to uniformity and durability of the product.

At the completion of the processing, the crankshaft must pass a final inspection which checks every possible point to assure meeting the standards established. The illustrations will indicate the degree of precision to which the crankshaft is subjected. Comparison will be made with previous methods used to indicate the progress in precision.

Crankshafts are wheeled into a completely enclosed and air-conditioned room.

Today's inspection methods have made possible interpretation of sizes and finishes to a degree not possible a few years ago. Setting the pace for continued processing development, they have themselves transformed the former crude and slow inspection into a highly scientific and systematic procedure. (Fig. 12.)

■ Fig. 12 – Modern inspection is a highly scientific and systematic procedure

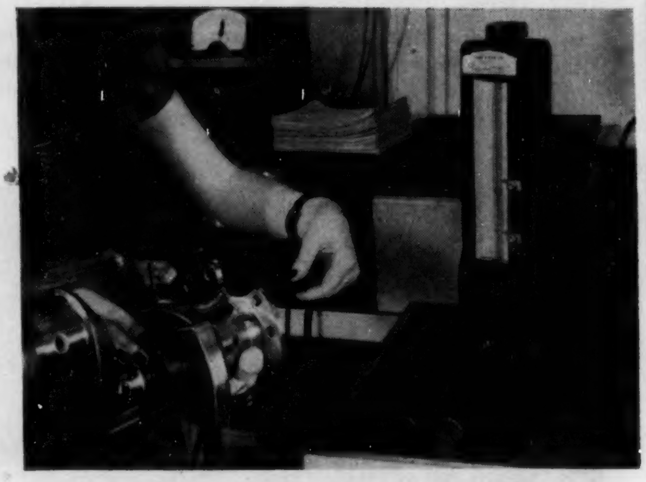
All operations are outlined according to stations and in the sequence in which they occur. Each stage involves precision checking to the highest standard of accuracy.

Fig. 13 shows the first stage, in which the crankshaft is hoisted into position for one of the final checks, which involve the use of a variety of solid-type gages. Standard and special plug, snap, bar, spacing, and flush pin gages check the accuracy of approximately 280 points.

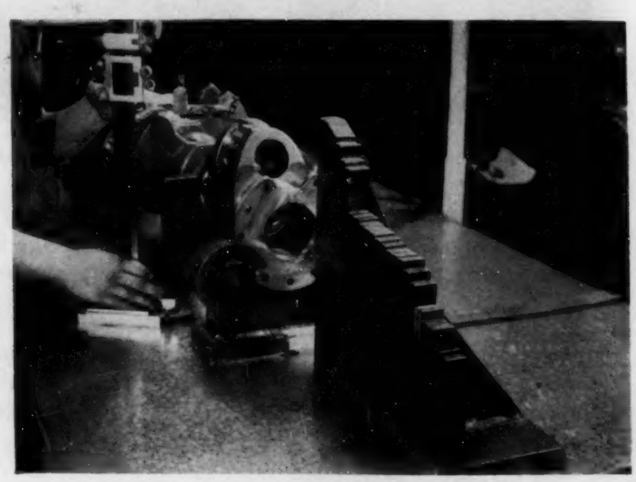
■ Fig. 13 – A variety of solid-type gages are used in the final check of all measurements

Fig. 14 shows the air gage checking of flange holes to the ± 0.0005 limit specified.

The latest development of precision checking methods for production inspection is the master steel block step gage (Fig. 15), designed to facilitate measurements that



■ Fig. 14 – Air gage checking of flange holes



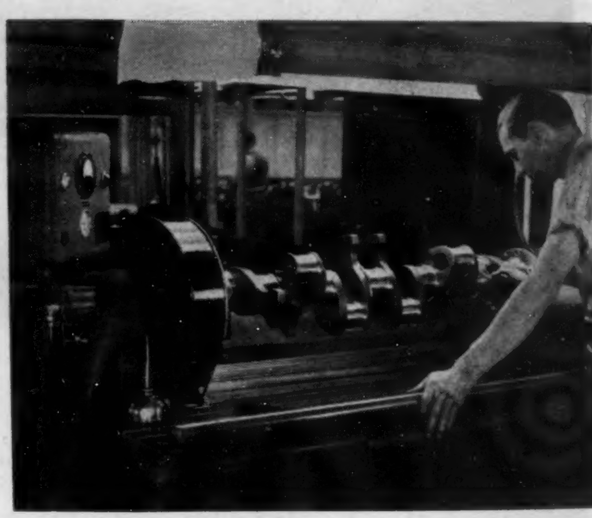
■ Fig. 15 – Checking with the master steel block gage – description of dimensions to be checked appears on step gage in lower right corner. In each case, the indicator is set to the step and the height is thus transferred to the crank

previously required the use of vernier-height indicators and surface plates. It is, in reality, a set of Jo blocks conveniently arranged according to the dimensions desired. A triple-step block is placed under the various crank throws to position them before inspection. The crankshaft is centered in fixed rollers and the entire set-up is mounted on a surface plate. A rapid and accurate check of a crankpin throw to ± 0.005 limits, concentricity of flange bolt holes within 0.002 total indicator reading, and checks totaling 173 separate points are systematically performed by properly instructed women inspectors.

Crank spacings are then checked using the steel column spacing gage as shown in Fig. 16. Mounted on the surface plate, the height of the spacings are set to an indicator and any variation is checked from "O." All main bearings crankpins, thrust faces, and so forth, are checked for length and spacing.

The purpose of the magnetic inspection (Fig. 17) is to segregate shafts with defects that are injurious to the strength, durability, or safety of the part involved. This is done by the direct-current wet method as specified by the Army and Navy. Particular attention is placed on the highly stressed areas, that is, fillets and their adjoining regions, and areas immediately surrounding all holes.

■ Fig. 16 – Steel column spacing gage being used to check crank spacings



■ Fig. 17 – Magnetic inspection being used to reveal defective crankshafts

Cracks or forging flaws, when characterized by open indications, are reasons for rejection. Detailed records are kept of all indications of any consequence. In many cases defects may be and are removed, but only when serviceability will not be impaired. Accepted cranks will have the magnetic inspection symbol etched on the side of the counterweight before passing to the next station.

This method of checking for defective crankshafts is relatively new, having been adapted to practical use within the past few years. Defects exposed previously were those detected during visual inspection by sounding and the kerosene and chalk methods.

Main and crankpin journals and bores receive a final check. Electric gages are set to masters – these check limits to be held to the 0.0005 specified. Bore gage indicators graduated in tenths pick out any irregularities in the bore circumference or its length. (Fig. 18.)

■ Fig. 18 – Electric gages, set to masters, are used to give a final check to main and crankpin journals and bores

Having completed its cycle, the crank will be subjected to a most rigid visual check, as shown in Fig. 19. Not even the slightest scratch, mar, or rust stain is acceptable. Before handling a crankshaft, each operator has taken precaution to prevent perspiration stains through the use of protective coatings for the hands.

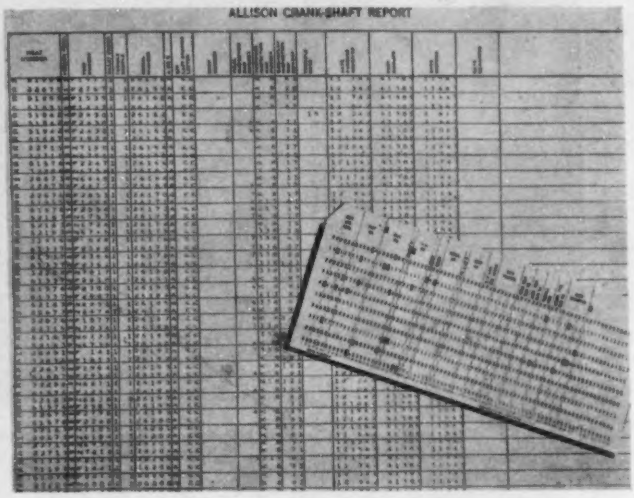
■ Fig. 19 – The crankshaft is finally subjected to a most rigid visual inspection, for the slightest scratch, mar, or rust stain will prevent the shaft from being acceptable

Although painstaking accuracy has become a law of inspection, each detail must be carefully recorded. Identification starts with the forge source and continues through each process step until the final assembly point has been recorded. Log sheets list every check and serve as positive guides for the inspector. Latest engineering change letter, shop, heat, and service numbers are stamped and then recorded for permanent record. (Fig. 20.)

The information from the log sheets is compiled on a permanent record. (Fig. 21.) Copies of records are kept at different locations for reference and as a precaution against loss. Any characteristics disclosed in service can be checked back against the manufacturing records to identify any shaft or any manufacturing process.

With the few facilities available years ago, a craftman's

■ Fig. 20 – Log sheets are kept for each crankshaft – detailing inspection and other pertinent data



■ Fig. 21 – Permanent records are kept of the data contained on the log sheets

interpretation of standards was measured and applied in terms of years of practical knowledge.

Dimensional control of tolerances and fits was limited through the use of slow, inaccurate gages and "rule of thumb" methods. Crankshaft finishes were accepted on a basis of men's ability to judge surface finishes by using his fingernail or by dragging a penny across the part.

Operating clearances were dependent upon an assembler's ability to scrape bearings to fit crankshaft contours. Bluing of mating parts provided the only means of checking contact.

Today, tolerances of 0.0005 are commonplace in our shops; parts processed in all corners of the world may be assembled with confidence. Desired running fits improve functional life of assemblies regardless of point of assembly.

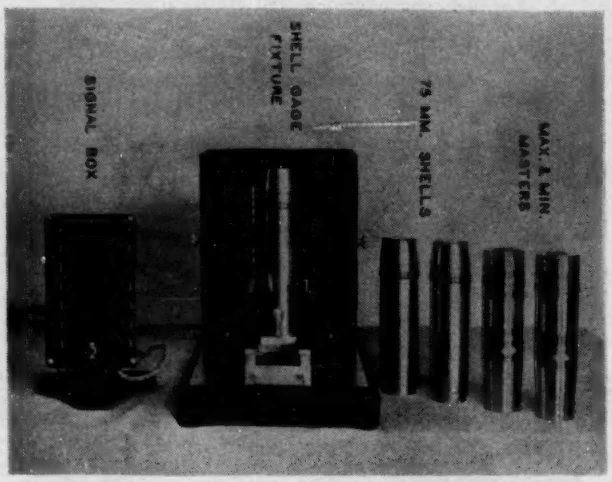
Finishes of today's crankshaft are held to within 2-

to 4-microinch range. Measuring devices indicate surface roughness.

The past 25 years have shown a noteworthy development in every branch of machine tool engineering, and in instruments for precise measurements. Their improvement has made possible the production of interchangeable parts in commercial quantities. As the means of measurement has improved, so has the degree of accuracy of production progressed. This has been indicated in following the various stages with the crankshaft. In addition to the methods of measurement indicated so far, there are many others which have been developed – and they all contribute to progress through permitting improved measurements of comparisons with standards.

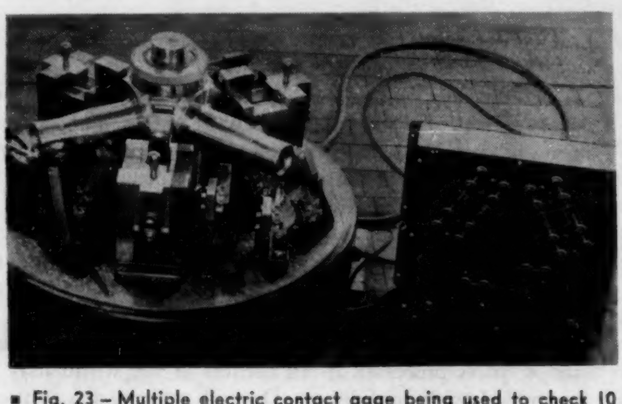
Typical of some of the outstanding methods are the following:

The gage illustrated in Fig. 22 is used in checking 75 mm projectiles. The signal box on the left contains a series of lights which indicate whether the shell is over or under the set limit. In this case, seven different positions on the shell are established by setting the shell gage fixture to the maximum and minimum masters shown at the right of the fixture itself. A multiple electric contact gage of this type can be utilized for accurate, high-speed checking, impossible only a few years ago.



■ Fig. 22 – Multiple electric contact gage being used to check seven positions on a 75 mm shell simultaneously

The multiple electric contact gage shown in Fig. 23 checks 10 diameters of an airplane propeller hub simultaneously. Group of lights is provided to show over or under limits.



■ Fig. 23 – Multiple electric contact gage being used to check 10 diameters on an airplane propeller hub simultaneously

Fig. 24 shows an excellent example of the manner in which the production operators are today provided with precision equipment to check their work. The internal-comparator gage shown is used for checking the connecting-rod wristpin hole. Taper and size are gaged at the machine where the hole is being ground. Inspection is adjacent to the operation itself, indicating the cooperative effort that must exist between the fabricating or processing department and the inspection department.

■ Fig. 24 – Internal-comparator gage being used for checking the connecting-rod wristpin hole

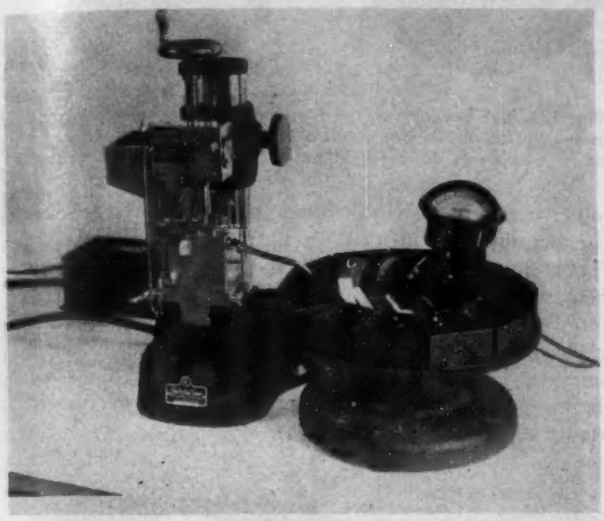
Another application of an internal comparator to modern methods is in the inspection of piston wristpin holes. (Fig. 25.) Inspectors' judgment factor has been reduced to practically nil, without detracting from the quality of the

■ Fig. 25 – Internal-comparator gage being used for checking the piston wristpin holes

finished product. Tolerances of 0.0005 or less are now common, giving further evidence of the scope of interchangeability made possible through the use of modern instruments.

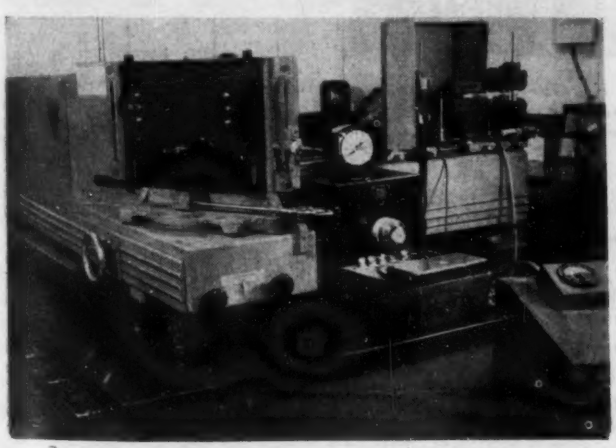
Basically, all electric comparators work on the same principle. Composed of a combination of electrical parts and mechanical adapters, their application varies widely.

Segregation of internal or external gaged parts is accomplished rapidly and accurately. The ball selector shown in Fig. 26 was designed for grading balls according to size. Operated by a mechanical trip lever, it drops a ball into gaging position. At the same time, the ball previously gaged is released and drops into the proper bin. Balls are selected to within 20 millionths of an inch variation.



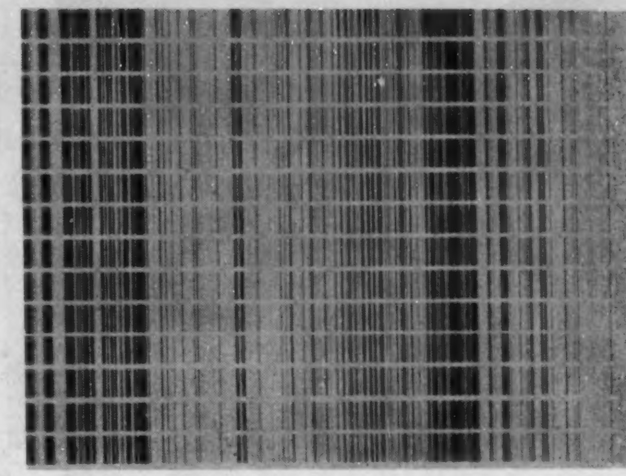
▪ Fig. 26 – Ball selector for grading balls according to size

The spectrograph (Fig. 27) is an instrument used for quickly estimating the chemical composition of material which, by chemical analysis, would take a long time and be difficult to figure. A sample to be checked is used for electrodes between which electric current is passed, sparking and burning the electrodes. The light from the spark is then passed through the instrument where it is broken up by a prism into lines of the spectrum and recorded on a photographic negative.



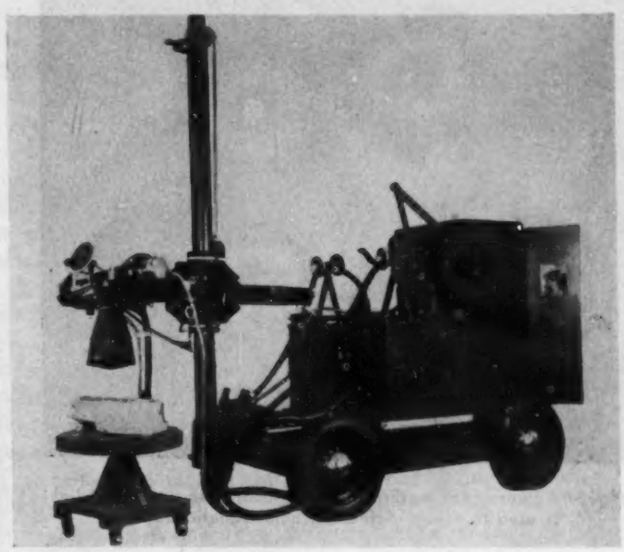
▪ Fig. 27 – The spectrograph is used to estimate chemical composition quickly

The lines recorded are compared with standards so as to afford not only qualitative but quantitative figures on the chemical composition. (Fig. 28.) About 4 hr would be taken in making a chemical analysis as compared with 20 min with the spectrograph.



▪ Fig. 28 – Spectrograms give both qualitative and quantitative figures on the chemical composition

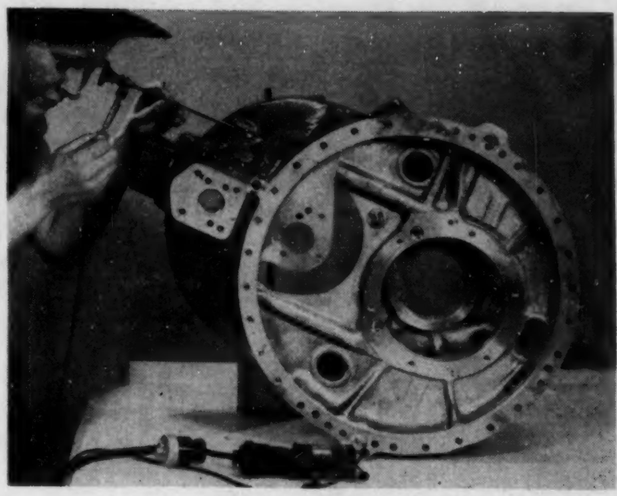
The X-ray utilizes an invisible emission of radiation which penetrates most materials of construction. (Fig. 29.) This radiation, commonly called X-ray, acts upon photographic emulsions. A photographic image is obtained by passing the X-rays through the material and then on to the photographic emulsion. The film, when developed and fixed, can be examined and discloses discontinuities, changes in section, and other characteristics of the material



▪ Fig. 29 – X-ray machines are used to discover hidden defects

examined. Hidden defects can be found without destroying the material.

The importance of cleanliness in assemblies and attention given to finished surfaces has extended the use of medical and surgical instruments to the field of modern industry. (Fig. 30.) Industrial telescopes constructed for any angle of vision are provided with lighting arrangements for inspecting most inaccessible corners and internal surfaces. Drilled oil passages, intersecting holes, and so forth, are scanned for material or machining imperfections which may interfere with proper functioning of the part.



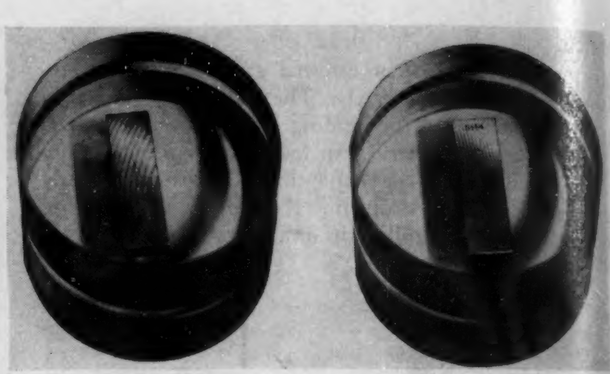
■ Fig. 30 – Industrial telescopes, constructed for any angle of vision, are provided with lighting arrangements for inspecting most inaccessible corners and internal surfaces

The ultraviolet light method is used to reveal defects such as cracks, voids, and porosity in nonmagnetic parts by using a fluorescent material in connection with ultraviolet light. Parts are immersed in a bath of fluorescent liquid. The surface is then cleaned and the fluorescent liquid exudes from the cracks and becomes visible under the ultraviolet light. (Fig. 31.)

■ Fig. 31 – Fluorescent material in connection with ultraviolet light is used to reveal defects in nonmagnetic parts

Optical flats are pieces of glass, quartz, or other transparent material ground and polished to extremely accurate flat surfaces. These surfaces, when placed on machined and polished metallic surfaces, cause colored rings of light to become visible, called interference bands, the size, number, and curvature of these bands enabling estimates of measurement down to millionths of an inch. Contrast in flatness of two surfaces is indicated by parallel straight lines in one and curved lines in the other. (Fig. 32.)

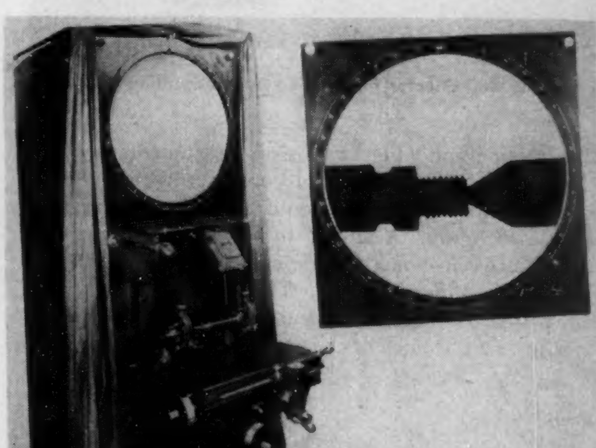
Shadowgraph, comparator, contour projector, and others use a light source, passing the light through a lens system, causing the rays to fall into parallel planes. These parallel rays are passed over the surface checked and then through



■ Fig. 32 – Part having machined and polished surfaces is placed between optical flats – from resulting interference bands, estimates of measurements down to millionths of an inch can be made

another lens system, which causes the rays to diverge so as to be magnified. (Fig. 33.)

These magnified rays are then projected on a screen where the image enables errors not visible to the naked eye to be seen and measured easily. In measuring contours such as threads, gear teeth, form tool cutters, and other geometric outlines, it permits taking advantage of full tolerances, since it shows clearly in enlarged detail any variations between parts checked.

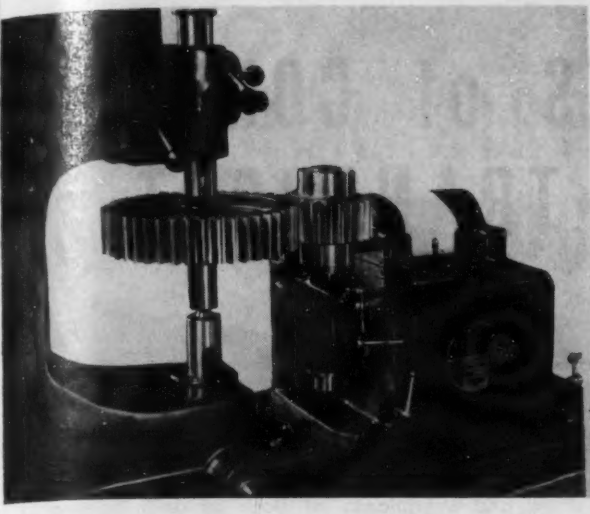


■ Fig. 33 – Shadowgraphs, comparators, and contour projectors are used to check contours of threads, gear teeth, form tool cutters, and other geometric outlines, to permit taking full advantage of tolerances

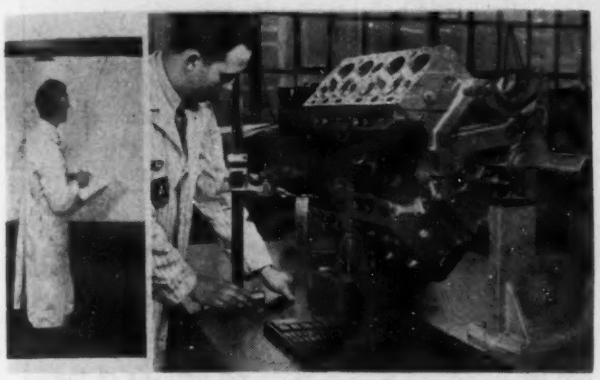
Used for production inspection of gears, the automatic machine shown in Fig. 34 will accurately record errors. A holder mounting a master gear is designed to be flexible; thus any errors in the gear being checked affect the center distance. Movements of the flexible holder are mechanically relayed to a pen which traces a line on a moving chart. Eccentricity, tooth shape, and tooth-spacing errors produce characteristics relating them to a straight line which can be readily identified. Both sides of the gear are checked simultaneously.

In checking a cylinder crankcase the old way, it was necessary to study the drawing for each dimension. All dimensions were then set up with Jo blocks and the part checked using a vernier height gage. To check a crankcase completely by this method required about 2 weeks' time. (Fig. 35.)

To devise a better method, all dimensions were num-



▪ Fig. 34 – Machine for automatically recording errors in a gear

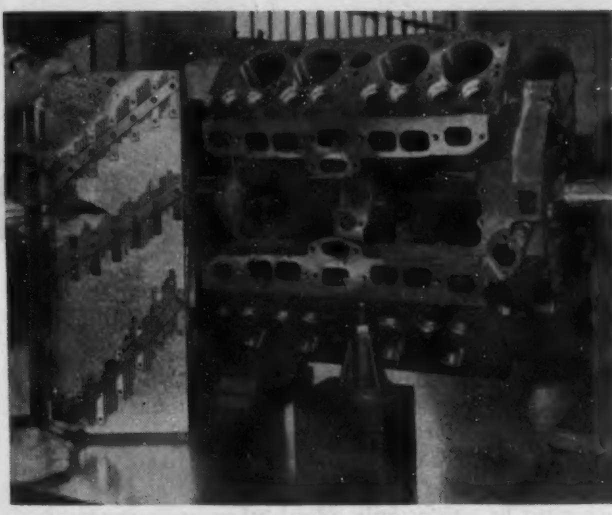


▪ Fig. 35 – Old way of checking crankcase with Johansson blocks and a vernier height gage

bered and charted on master sheets. Each dimension was assigned to a color, indicating vertical, horizontal, and angular positions of the crankcase. Master sheet colors and numbers match step pins mounted on a master block. To check a crankcase it is only necessary to transfer checking points from the block to the crankcase. (Fig. 36.)

By this method only 8 hr are required to complete a crankcase check contrasted with 2 weeks' time spent the old way.

The development of precision instruments has made it possible to train inexperienced help, women, and previously inexperienced older men from other trades, in the precision requirements of mass production to compensate for the drain upon experienced personnel caused by the war itself. Today because of these instruments we train inexperienced personnel in terms of weeks, whereas with our cruder instruments of the past we trained personnel in terms of years; and I feel that it is obvious, measured in terms of the products we build today – quantity, quality, cost – we build better products and man enjoys a better living as a result. Each builds for the other. It is like a repetitive cycle yet constantly moving forward. Ten years, five years, even one year from now our instruments, our tools, and our products of today, in which we take such pride, will be as outmoded as those of 25 years ago as we look at them today. This is what I mean by progress in precision – past, present, and future.



▪ Fig. 36 – New method of checking crankcase is to use master sheet colors and numbers, which match step pins mounted on a master block – inspection consists of transferring checking points from the block to the crankcase

Current Trends in Lubrication

SOME of the current trends are:

1. Compounded oils are becoming more common. Oils containing compounding materials frequently have unusual properties not found in any uncompounded oils, and in this respect they may be considered analogous to alloy steels. Research work on compounded oils did not become a major activity of the petroleum industry until the 1930's. From 1930 to 1940, special lubricants for high-speed diesel engines were developed and compounded automotive oils started to become popular. Today, when a difficult lubrication problem arises, the usual solution is to prepare a tailor-made product composed of the best oil stock and the most suitable compound.

2. Lubrication specifications are being made more functional. In the past, these specifications contained such tests as gravity and viscosity. It was found that these tests were inadequate for indicating the quality of a compounded oil, and many modern specifications contain test requirements very much like the service for which the products are intended.

3. Lubricants for the military service are being simplified with respect to numbers of grades and types. It is believed this trend is primarily a wartime measure to facilitate transportation of products to remote places and to eliminate improper fillings in the field. It is doubtful that this trend will have much effect on commercial applications.

4. Probably the most significant trend in lubrication is that of continuous and definite improvement in the quality of all products. This improvement has been continuous for a number of years, and there appears to be no reason to believe that it will not continue. Therefore, even better lubricants can be expected in the future.

Excerpts from the paper of the same title by F. W. Kavanagh, Standard Oil Co. of Calif., presented at the SAE West Coast Transportation and Maintenance Meeting, San Francisco, Aug. 20, 1943.

DEFICIENCIES of CONVERTED PA for CARGO TRANSPORT and OP

ONLY 10 months have elapsed since the airlines of the United States were called upon to begin the transportation of cargo on a large scale for the military services. The volume of cargo so far flown and its rate of daily increase have presented cargo handling and transporting problems which must be satisfactorily solved before air cargo transportation can become a routine operation.

Expedient solutions are meeting the present emergency but fundamental thinking and reasoning must be followed to derive specifications applying to cargo airplanes in order to establish the transportation of air express and air freight as a commercial business at the termination of the war.

Flying has become an accepted means of transportation by the commercial traveler as well as the vacationist, but the carriage of goods by air has lagged primarily because of the lack of suitable flying equipment and handling facilities. Furthermore, the relatively small amount of space available for air cargo has been indirectly the cause for the existing high tariffs which, in most instances, have been higher per pound per mile than comparative passenger rates.

When it became obvious at the beginning of this war that speed of transportation for critical military material was essential, it was logical that available passenger equipment in use, or already designed and ready to be produced, should be quickly altered to meet this immediate demand.

The result was that airline equipment was pressed into cargo operating service literally overnight and speedily converted for such service by the removal of all passenger accommodations, strengthening the floors with additional covering, and changing the radio equipment to meet military requirements.

This conversion was effective, although rather inefficient in so far as maximum ratio of useful load to gross weight is concerned, but this was partly offset by lifting commercial weight restrictions and increasing take-off weights in the order of 20% with surprisingly safe and reliable results, which, incidentally, emphasize the conservativeness of commercial equipment maximum operating weight ratings.

The standard Lockheed Lodestar, Douglas DC-2 and DC-3 transport airplanes were thus modified and are doing an excellent job. Later on, production of the C-47, which is basically a DC-3 cargo version having a large side loading door; the C-46, a modification of the original 36-passenger Curtiss-Wright CW-20 commercial transport airplane; and the C-54, a conversion of the production DC-4 airplane, swelled the ranks of cargo airplanes to expedite movements of military personnel and material. A more recent modification was that of the Consolidated B-24 four-engine bomber to long-range cargo carrier and known as the C-87.

[This paper was presented at the SAE Air Cargo Engineering Meeting, Chicago, Dec. 8, 1942.]

SHORTCOMINGS of the converted passenger plane are at least 10 in number, says Mr. Froesch, classifying them as follows: (1) floor slope and irregularity of floor at door causing concentration of load at that point; (2) floors too weak, requiring reinforcing; (3) doors too narrow for entrance of bulky loads; (4) no anchorage for load fasteners; (5) no provisions for cargo handler station; (6) hard to distribute load so as to give a satisfactory center-of-gravity location; (7) lavatory in the wrong place; (8) insufficient fire extinguisher protection; (9) door sill heights too variable; (10) circular or oval fuselage shape, which cannot be used effectively.

A big problem is to get the rate down to the point where repeat business will follow. Eventually this might mean a 10 to 12¢ per mile rate under proper designing, Mr. Froesch declares. But, he says, size and capacity of the cargo plane cannot be predicted until a thorough analysis of the air express and freight market has been made.

A density-volume ratio of 8 to 9 lb per cu ft can be used, Mr. Froesch says, as a design criterion in figuring size of compartment. He points out heating and ventilation aspects of the cargo compartment and explains desirability of having a small compartment for protection of valuables close to the cockpit. The airplane should also be designed so that preventive maintenance and service methods can be best applied.

THE AUTHOR: CHARLES FROESCH (M '16), chief engineer for Eastern Airlines, Inc., has witnessed the effect on aeronautics of two world conflicts. An inspector of aircraft, aircraft engines and aeronautical design during World War I, he deserted the field temporarily to become bus engineer, and later took charge of gas-electric bus design for Mack Trucks, Inc. He returned to aviation via the Fokker Aircraft Corp., serving as project engineer, assistant chief engineer and general service manager. His present post he has held since 1934.

While the following discussion covers the deficiencies of all these airplanes as applied to air cargo operation, it must be remembered that none of them were designed strictly as cargo carriers or for conversion to air cargo transportation and, therefore, none of these criticisms and shortcomings can or do reflect, in any manner, on the equipment manufacturers themselves.

PASSENGER AIRPLANES
OPERATING REQUIREMENTS

by **CHARLES FROESCH**
Chief Engineer, Eastern Air Lines, Inc.

▪ Fig. 1

▪ Fig. 2

In the first place, no one knew just what a cargo airplane ought to be, and with limited airline experience in cargo operation, much difference of opinion still prevails on the characteristics and specifications involved in the design of this type of airplane.

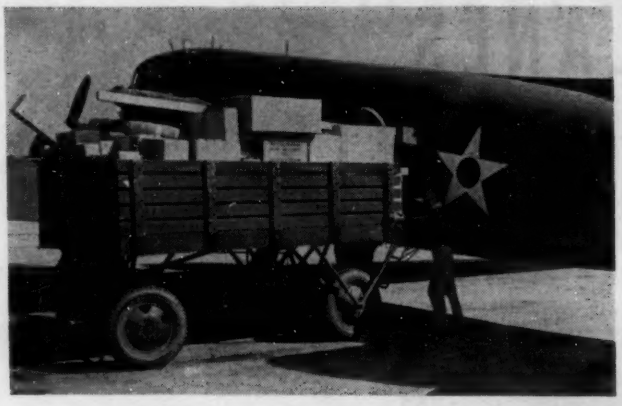
However, we are fortunate to be in the same position as the beginner photographer who is proverbially taught how to take bad pictures in order to learn how to take good ones.

Let us, therefore, analyze the deficiencies of these converted airliners and outline suggested modifications for future equipment in the light of our limited operating experience.

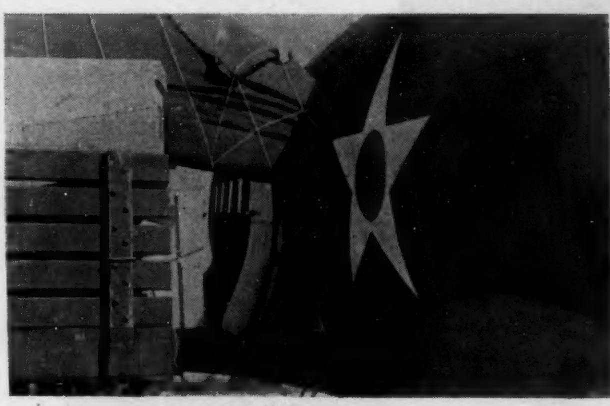
First of all, we find that the conventional type of landing gear renders loading of the cargo compartment with heavy cargo pieces difficult because of the floor gradient. This is aggravated by the lack of anchor points for hoist attachments to help pull heavy items at the entrance door or in the cabin. Also, in order to have a necessarily horizontal floor in the region of the loading door, the main floor makes an angle with it and causes concentrated loading when handling heavy, long pieces of cargo, with resultant floor damage.

Second, we find that the standard passenger cabin floors are too weak to stand concentrated loads as well as the rough usage to which they are now subjected as cargo carriers. This has demanded their reinforcement by either strengthening the present floor structure or covering it with an additional floor. This supplemental floor consisted generally of plywood sheeting $\frac{3}{8}$ to $\frac{3}{4}$ in. thick and has been found quite serviceable although it substantially increased the empty weight.

Third, we find that in the converted airliners, the cabin entrance door is too narrow and too low to take the bulkier cargo pieces (Figs. 1 and 2). This restriction seriously impedes efficient loading and unloading and limits the size of cargo pieces which can be handled (Fig. 3). This is an important item when it is considered that the size of cargo



■ Fig. 3



■ Fig. 4

pieces may commonly run from an article so small as to require wrapping its shipping label around it instead of pasting it, to automotive units measuring almost 13 ft in length, 6½ ft in width, 4½ ft in height and weighing 2500 lb. Occasional loads have greater dimensions and/or weight. It would be desirable to have an additional door, located opposite, to enable simultaneous loading and unloading and thus save time on the ground, particularly when one or two pieces of a load are difficult to handle, due either to size or weight (Fig. 4).

Fourth, we find that when we have placed the cargo within the airplane in its proper location, no means are available to anchor the tie-down cables, ropes, or other devices used and a makeshift installation must be provided.

Each type of cargo presents a somewhat different problem in tie-down procedure. Methods of tie-down vary with individual airline interpretation due to lack of basic provision in the airplanes. Some airplanes have ring tie-down fasteners attached to the floor frames, others have side rails, and still others employ a combination of both means. The most common practice is to use ropes tied in an indiscriminate manner, usually following the fancy of the handler lashing the cargo (Figs. 5 and 6). Nets are used but need frequent replacement.

Many ideas are being tried and each particular type has its own merits and drawbacks. No matter what tie-down means are successfully developed, their satisfactory application will depend primarily on the proper training of the loading personnel in their use.

Fifth, we find that no provisions are readily available for a cargo handler station.

Sixth, we find a lack of simple and concise information to permit the loading personnel to distribute the cargo load accurately within the prescribed balance limits. In addition, it is often difficult to place heavy pieces within the cabin and still retain a center-of-gravity location which will not seriously affect longitudinal stability as well as take-off and landing characteristics.

For instance, when a load of 6000 lb is carried in the C-49, its center of gravity must be at a point roughly 28% of the cabin length measured from the forward cabin wall. This seriously limits the available placement volume if the cargo is bulky.

The arrangement of cargo is at best a process requiring intelligent action in view of the variable weight, size, and type of cargo pieces, besides the fact that such cargo may be bound for several destinations requiring placement in the compartment to allow sequence unloading.

Loading requires the coordinated effort of the plane crew, ground operations, loading personnel, and, when in flight, communications to keep station delays due to loading and unloading at a minimum.

Seventh, we find that the lavatory compartment is in the wrong place as it is often difficult, in flight, to reach it from the pilot's cockpit with an airplane fully loaded with bulky material of low density.

Eighth, we find that no provisions are available for fire detecting and extinguishing in the passenger cabin con-

■ Fig. 5

■ Fig. 6

vented to cargo compartment as hand fire extinguishers are the only requirements for passenger operation.

Ninth, we find that each type of airplane has a different doorsill to ground height, necessitating lifting or lowering the cargo when loading from a standard motor truck platform height. This has resulted in serious delays, trip cancellations, and inability to operate on schedule. For instance, the doorsill height from the ground is but 3 ft 6 in. for the C-87, whereas it is 9 ft 8 in. for the C-54, or a difference of 6 ft 2 in. between the two.

It has necessitated the use of makeshift devices, such as loading ramps (Fig. 7), to enable transfer from standard truck platform height.

Tenth, we find that the cross-section of the fuselage, usually of circular or oval shape, does not permit maximum space utilization.

These are 10 primary deficiencies found in the operation of converted passenger transport airplanes for air cargo service as well as in airplanes originally designed for passenger operation and factory-modernized for cargo carrying.

While our primary purpose at the present time is to win the war, we must, nevertheless, look deeper and make plans for the future, or at least try to conceive what air transport operations are likely to be afterwards. Naturally, whatever changes are made by following this line of thought will also apply to military cargo airplanes with the exception, perhaps, of speed, which for military purposes must remain more important than operating cost.

In thinking of the progress of air transportation and its place in our national system, we must try to find its proper relationship with other available surface means. Obviously, we cannot hope to fly every traveler and carry every type of merchandise with maximum economy and must, therefore, segregate passengers and cargo according to purpose and distance of travel.

Before we can tentatively set down operating requirements for airplanes designed basically for commercial air cargo operation, we must first determine just what are the fundamental laws which apply to the transportation of air cargo.

The principal service the air cargo operator has to offer to the shipper is the speedy and safe transportation of his products. He must furnish this service at a sufficiently low tariff to attract and continue to carry a large volume of traffic. Air express volume has been built in the past primarily on the demand for emergency and perishable shipments which could bear a rate premium and as such can never grow to a substantial size.

Our first consideration, therefore, should be lowest possible operating cost per ton-mile. This is a necessity if we desire to obtain a large volume of business and compete with other means of express and freight transport which now have substantially lower rates per ton-mile from door to door than we have been able to obtain with the airplane to date.

First-class rail express is moved at a cost to the shipper at an average of 10 to 12¢ per ton-mile and if this rate can be met, including pickup and delivery cost, there should be a large volume of business available for movement by air and including most commodities which now go by rail express.

We certainly cannot hope or expect to supplant existing means of surface transport, as at one time or another in the course of modern processing or fabrication of commodities or articles, practically all such means are utilized.

■ Fig. 7

The airplane will probably limit its service to the transportation of merchandise representing either a high density or volume cost such as expensive tools and vacuum tubes, thus becoming intermediate means of carrying finished products from factory to consumer with maximum expediency.

A 10 to 12¢ per ton-mile rate is not impossible if we judiciously interpret and apply our present knowledge of aerodynamics, improvements in structures, and larger powerplants to the design of strictly cargo-carrying airplanes.

It is true that mileage from point to point is invariably lower by air and that the elimination of heavy packaging decreases cost of cargo preparation and promotes efficient utilization of the airplane, but still these advantages come far short of overcoming the differential between air and rail express tariffs.

We can possibly go further by exploring the field of l.c.l. freight which might possibly be handled as second-class or deferred express, bearing a lower rate and used to increase the average load factor, thus reducing average ton-miles/hr cost.

At a later date we may conceive the use of cargo glider trains with take-off and landing strips alongside factories simulating present railroad sidings. However, this discussion is limited to operating requirements of cargo airplanes only.

A cargo airplane should be a vehicle to carry merchandise not only at the lowest possible cost, but at the highest permissible speed. High speed is essential but if it is to be obtained purely by a reduction in cruising power loading, operating costs will be higher, since speed varies as the cube root of the horsepower. This being the case, tariffs would have to be raised and volume obtainable lowered.

For a given airplane, designed for a specific gross weight, and having a given wing loading, there is a power loading which is most economical. In other words, there is a balance between maximum payload and lowest cruising horsepower on one hand, and low power loading with reduced payload on the other.

This can be readily determined with a given airplane by calculating its transportation efficiency factor expressed in ton-miles/hp-hr for several horsepower values, that is:

$$\frac{P_l \times V_c}{hp_c}$$

where:

$$P_l = \text{payload in tons}$$
$$V_c = \text{cruising speed in mph}$$
$$hp_c = \text{cruising horsepower}$$

The over-all efficiency factor of the airplane can be obtained by multiplying the structural efficiency expressed by $\frac{\text{operational useful load}}{\text{gross weight}}$ by the aerodynamic efficiency as expressed by $\frac{\text{cruising speed}}{\text{horsepower}}$ and plotting against range.

Obviously, there is nothing to be gained if, due to high speed, the cargo must wait several hours to be unloaded at destination, such as might occur with a load of perishables having left the field of origin the evening before and arrived at destination several hours ahead of customary delivery time.

I fear to tread on the question of airplane size, except to mention that it must be remembered that larger airplanes permit higher ratios of useful load to gross weight and also greater range for a given load, as many airplane components and detail weights do not increase proportionately to the gross weight. Airplane size at present is limited by available engines, although this existing restriction may be reversed with the conception of large airplanes requiring several years to design and fabricate experimental prototypes. As a matter of fact, engines will need to be conceived and developed to permit lower fuel cost per ton-mile and more miles per gallon or else air cargo expansion may find itself seriously curtailed, especially for long-range operation.

Cargo airplane size and capacity cannot be predicted until a thorough analysis of the air express and freight market has been made. All we know is that air express is here to stay and that its volume will depend on the cost of the service to the shipper. Also, that air freight will follow in the wake of any rate reduction which can be made to render its transportation competitive with other means.

Obviously, the sales price of any cargo airplane must be as low as possible because, on the basis of modern experience, depreciation amounts to, roughly, 10% of direct flight costs.

Type

Generally speaking, the type of cargo airplane which appears most suitable points to a high-wing bimotor with tricycle gear (Fig. 8). For larger capacities and longer range, a four-engine low-wing plane might be considered superior.

Because of its simplicity, the bimotor type is more economical and meets the most important operating requirements for the following reasons:

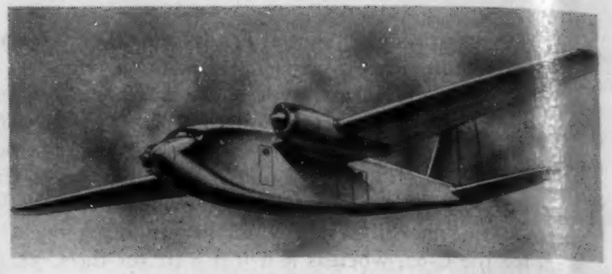


Fig. 8

1. It has the simplest powerplant combination and meets the requirement for safe operation and reliability with least servicing and maintenance. Single-engine performance must be sufficient to maintain a safe altitude over the highest obstacle to be negotiated.

2. A high-wing combination permits a low fuselage and, therefore, low door height to ground, besides having ample wing-to-ground clearance to allow freedom of movement for ground vehicles without danger of hitting the wing (Fig. 9).

Furthermore, large propellers may make a high-wing or mid-wing location almost mandatory for any cargo airplane up to a given size.

3. The use of a tricycle landing gear is necessary to obtain a level loading floor and ease of loading and unloading.

In practically all landings, the pilot usually flares the airplane more than is necessary, with resultant higher vertical velocity upon impact on the runway unless lift is retained by high landing speed.

From an operational viewpoint, the elimination of the landing flareout necessary with the conventional type of landing gear is an advantage, as with instrument landing in bad weather the airplane must be virtually flown to the ground.

One of the basic characteristics of cargo airplanes should be the largest possible center-of-gravity travel to eliminate critical load distribution.

Cargo Compartment

The cross-section of the cargo compartment should be square or rectangular in section for maximum space utiliza-

Fig. 9

tion and cargo hold-down anchor location simplification. There should be no sharp corners which might be a source of personnel hazard, besides being dirt collectors. The minimum height of the compartment should be 6 ft 6 in., to allow walking throughout its entire length erectly.

There should be at least one large loading door on the left side, together with a smaller-size one for crew entrance and exit at the front end of the same side. Preferably, and I repeat what I mentioned earlier, another large loading door should be located on the opposite side somewhat ahead of the left loading door to enable loading and unloading at the same time. The relative location of these doors lengthwise to the cargo compartment could readily be determined from actual experience with a mockup test.

The doors should be designed to present minimum obstruction to loading and unloading. Flush-mounted sliding doors would best meet this requirement.

Cargo compartment floors should be on the same level to expedite loading and eliminate steps as a source of accidents. Such flooring should be of sufficient strength to take a high load concentration and the entrance door region should be designed to take a load concentration equal to the heaviest piece expected to be carried. This may require a good guess, as about 90% of all express packages are shipped in corrugated boxes or light wooden crates which can be carried by one man. However, the remaining 10% may include aircraft engines weighing up to 3500 or 4000 lb as well as other pieces of heavy machinery which have been carried in the past as emergency shipments.

From the economic point of view, it does not appear logical to increase the empty weight of every cargo airplane of a fleet to handle this 10% of cargo and the solution may be that only a few such cargo airplanes will be sufficiently strengthened to take what is virtually heavy freight.

Cargo space volume for any given airplane size should be based on an average express or freight density.

Rail freight averages 32 lb per cu ft and rail express about 12 lb per cu ft, while airmail averages 9 lb per cu ft. If we assume that improvements in packaging can reduce express tare 20%, we arrive at an air express density of approximately 9½ lb per cu ft.

Allowing for unusable space plus a certain amount of space in the region of the loading door which should probably be kept clear at all times, we can set down a density volume ratio of 8 to 9 lb per cu ft as design criterion in determining the size of the cargo compartment.

Cargo stowage means should be light and efficient, suitable cargo tie-down devices should be provided flush-mounted and built integrally with the floor and side wall structures. Provisions should be made for a flexible segregation of the cargo compartment space into bins and sections, such as may be obtained by the use of movable stanchions with flexible partitions such as chain doors, which are light (Fig. 10). The side walls up to a minimum height of 36 in. from the floor should be protected by means of suitable removable panels, fabricated perhaps from laminated plastic having varying thickness from the ground to their maximum height to secure least weight for maximum protection.

Only sufficient windows should be incorporated to give satisfactory light and these could be portholes. Cargo compartment lighting must be protected against damage. Such lighting should be adequate as much work will be done at night.

■ Fig. 10

■ Heating and Ventilation

The entire cargo compartment should be well ventilated without being drafty. Heating should be provided in the design to maintain a 50 F temperature, which is necessary when perishables such as flowers, fruits, and animals are to be transported.

It will be necessary when operating in cold climates to provide temperature insulation. Such insulation should be made readily removable for two reasons: First, weight saving when it is not needed, and second, to allow periodic cleaning.

It is not believed that the cargo compartment, as a whole, needs to be soundproofed, although such soundproofing should be made available in sections in case it should be required.

Provisions should be made for air conditioning and possibly refrigerating.

■ Cockpit

Cockpit arrangement should include space for cargo handler station (Fig. 11). The cargo handler's duties would be to keep track of all cargo pieces, check and prepare cargo manifest, check destination of cargo, supervise loading and unloading, check the center-of-gravity location of the cargo for balance, as well as any other "in flight" duties which may be required.

A small cargo compartment should be provided and readily accessible from the cockpit for the safekeeping of valuables. This compartment should be close to the cargo handler's station as it would be his responsibility to care for such valuables.

■ Miscellaneous

Among the miscellaneous operating requirements can be cited adequate fire protection for both powerplant and cargo. For the latter, it is believed that a smoke detector should be a prerequisite as well as quick-acting extinguishing means.

Refueling and reoiling should be accomplished from the under-wing surface to eliminate the necessity of climbing over the top of the wing, which in the past has always caused substantial distortion of the wing leading edge, has

been the source of accident hazard, besides incurring additional labor time.

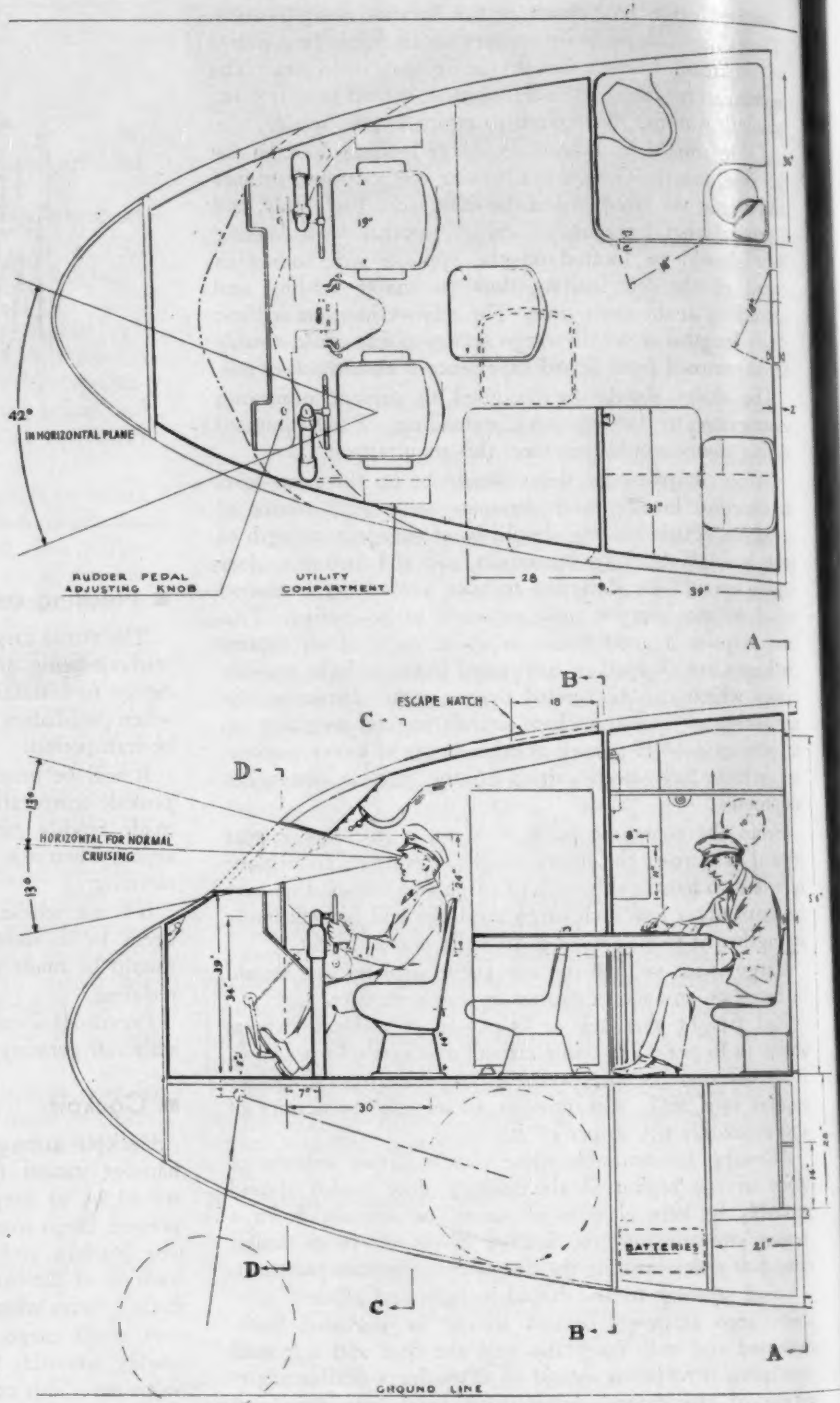
Obviously, powerplant assemblies must be designed so as to be quickly changed. This is a most desirable feature which has been lacking on all transport airplanes designed to date.

In general, reliability must govern detail design, which should consider adequate bearing surfaces, ease of lubrication and self-lubrication for inaccessible places, adequate brakes, readily accessible and removable engine cowling and inspection doors, bearing in mind that maximum utilization of equipment is one of the requirements for low operating cost.

In other words, the airplane should be designed so preventive maintenance and service methods can be best applied.

Cabin pressurization offers an increase in speed at high altitude but it is questionable whether the increase in structural weight and initial cost, as well as additional service and maintenance costs, will warrant its application to cargo airplanes, at least for the time being.

In order to simplify the ground handling of the cargo prior to loading or unloading, and gain a saving of ground time and labor, I would like to enter a plea for standardization of cargo loading door to ground height, possibly door size, cargo tie-down equipment within the cabin itself, and containers for the collective handling of small shipments, as it is not believed possible to standardize to any great extent on package size, except perhaps on the type and materials used in packaging in order to reduce tare, which represents a high percentage of the total weight of the article being shipped. This is an important item which needs emphasis and will require an intelligent and imaginative study when it is considered that at present approximately 30% of the weight of express shipments is packaging.



■ Fig. 11

The development of air cargo will serve to accelerate the distribution of national and international wealth and promote similar commerce. It will allow greater enjoyment of the material wealth of the earth and, it is hoped, will also serve to eliminate intercontinental friction which since the world began has been the source of destructive warfare. This may be an idealistic goal, but it is my belief that the airplane, as a vehicle of speedy transport, can make it real.

WARTIME MAINTENANCE of RINGS, PISTONS and CYLINDERS

by LEE DOTY

Koppers Co., American Hammered Piston Ring Division

IN this paper, I will discuss wartime maintenance problems in connection with rings, pistons, and cylinders of internal-combustion engines only as they are affected by the unusual operating conditions brought about by the present world situation. I am not in a position to discuss lubricating oils, except in a very general way, and will therefore allude to them only in general terms, where it is necessary, in connection with a discussion of the subject.

Briefly, during these unusual times, motor transportation may be broken down into two groups, with the first major subdivision of *passenger cars,* with a loose breakdown of those operating under A and B gasoline rations and those operating under C gasoline rations. The second main division would be *trucks and buses* and similar types of equipment, subdivided into those on short hauls or urban runs and those on long hauls.

We all know that the war has had a tremendous effect on our methods of operation, and there has been many a headache and much hair lost in trying to find the answer to the various problems as they present themselves. We will deal only with those factors which are most likely to cause difficulty with the rings, pistons, and cylinders in an engine.

Probably the most important of these conditions is the inability to procure gasolines of the same high standards

[This paper was presented at a meeting of the Baltimore Section of the SAE, May 13, 1943.]

OPERATION of internal-combustion engines under wartime conditions has created some new problems of maintenance and operation and has aggravated many of the old ones. Mr. Doty discusses this subject as it is related to rings, pistons, and cylinders.

1. Broken rings and broken ring lands between the first and second ring grooves, and occasionally between the second and third ring grooves.

2. Formation of excessive quantities of sludge in the crankcase. Somewhat lighter oil (of the detergent type if it is available), and more frequent oil changes will alleviate this condition.

3. Seemingly more rapid formation of lacquer and varnish deposits throughout the engine, particularly on the pistons and the cylinder walls.

Mr. Doty also discusses blowby and scuffing or scoring. Care in pulling down bolts when assembling an engine that has been rebored or re-ringed, especially the use of the torque wrench, is most important in minimizing distortion of the cylinder.

Reboring to too large an oversize or the excessive formation of sludge in the crankcase can lead to blowby. Piston scuffing or scoring is brought about only by the breakdown of the film of lubricating oil between the piston and its rings and the cylinder wall.

In conclusion, Mr. Doty states that two general precautions should be taken to secure maximum mileage betwen major overhauls at the lowest possible cost and with the greatest possible efficiency: *avoid detonation* and use *proper maintenance.*

★ ★ ★

THE AUTHOR: LEONIDAS DOTY, JR. (J '39) has been working on automobile engine problems for 20 years. As engineer in charge of the Automotive Division, American Hammered Piston Ring Division, Koppers Co., he determines what piston-ring set-ups should be made, takes care of all service matters in connection with the use of rings in the field. and personally contacts jobbers. Mr. Doty joined American Hammered in 1936. He worked in the shop for a time and was then made assistant in his present department prior to becoming its chief engineer. He is a graduate of the Baltimore Polytechnic Institute, and studied at Johns Hopkins University.

as those to which we were accustomed in peacetime. I am informed on reliable authority that the octane rating of the average so-called straight gasoline seems to be averaging about 71 to 72; and blended or leaded gasolines, as the case may be, seem to be averaging about 78 to 80. This is a comparatively small drop, on the face of things, but, unfortunately, other characteristics of the gasolines have suffered along with the drop in octane rating. The use of these fuels in engines which were designed and adjusted to operate on the high-grade, peacetime fuels, without suitable adjustments, and in some few cases alterations, will invariably result in trouble.

Wartime Operating Difficulties

With the formation of the tremendous army which this country is building, it is only natural that we should lose from the industry some of the most valuable maintenance men, and this has brought about a condition whereby the car owner, who goes to his neighborhood garage, and the fleet operator, who maintains his own repair shop, find it almost impossible to maintain the vehicles properly. The lack of this proper maintenance plus the inferior fuels, which have been mentioned, and several factors which will be covered later, can result in only one thing, and that is less satisfactory and economical operation of the engine.

To the car owner driving on the C ration who must have his car for transportation to and from work six or seven days a week, particularly when he is employed in an outlying defense plant, and the fleet operator who is already overburdened with work, the loss of the vehicle for even a day is a serious matter; but the scarcity of replacement parts frequently causes these vehicles to be laid up for weeks at a time, throwing an extra heavy burden on those vehicles which are still operating. Substitute materials, which admittedly are inferior to those intended for use in the engine, are also tending to throw more emphasis upon proper maintenance when the manpower to take care of this maintenance is not available.

The fleet operator, in addition to the various factors mentioned above, is faced with two additional very serious problems. One, the serious overloading of the vehicles because of the present wartime business, and the other, the inferior type of operator whom it is often necessary to use on his vehicles. The fact that jobs are plentiful in most localities has made it necessary for the average operator to coddle to some extent what personnel he has, as they are almost irreplaceable, all of which throws an extra burden on the vehicles and the component parts of these vehicles.

The owners and operators of passenger-car vehicles, in addition to the gasoline and maintenance difficulty, if anything are receiving even a poorer grade of gasoline on the whole than is the fleet operator, and in addition, many of the high-grade lubricating oils which were available to the passenger car operator can no longer be procured. The introduction of gasoline rationing with the reduction in overall mileage, and the Federal law limiting speeds to 35 mph, are throwing additional burdens on the engines, and especially on the lubricating oils. We will touch on this subject in more detail later.

The effect of these various factors on the engines of all vehicles has been to decrease generally the amount of satisfactory mileage between major overhauls, and in many cases completely to destroy parts, such as pistons, which under normal operation could have been used for many additional miles. Hard starting, particularly in cold weather, coupled with fuels which do not completely burn under certain conditions, results in excessive wear and increased oil and gas consumption.

Functions of the Piston Ring

Before we go into the question of the various problems that have been brought about by present-day operating conditions, it may be well, by way of review, to state just what a piston ring is and what functions it is expected to perform. A piston ring is, generally speaking, a bearing, constructed from a very high-grade gray-iron individual casting, designed so that it has what is commonly called tension, either built into the ring or derived from supplemental springs placed between the ring and the bottom of the ring groove, so as to force the ring to follow very closely the cylinder wall. Its purpose is to form a seal between the piston and the cylinder wall to prevent the passage of combustion gases or excessive quantities of lubricating oil from passing between the piston and the cylinder wall, and at the same time provide adequate lubrication of the piston and ring in the cylinder at all times throughout the entire length of the piston travel.

Contrary to general belief, piston rings, if closed up in a flexible band, are not round, as a round piston ring will not perform at all in an internal-combustion engine, but they are so designed that they exert uniform pressure against the cylinder wall at all points. They may be divided, in general, into two different types – *blowby control rings* and *oil control rings* – but the two types are definitely related in their performance, one depending upon the other, and there is a definite relationship between blowby and oil consumption.

Breakage of Rings and Lands

One of the conditions which has been most frequently brought to our attention in the past few months by persons interested in practically all types of internal-combustion engines, is the problem of either broken rings or broken ring lands between the first and second ring grooves and occasionally between the second and third ring grooves. When we pull a piston from an engine and have it in our hand, and find that either the ring or the ring land is broken, it is quite probable that we will jump at conclusions as to the actual cause of the trouble.

First, there are certain factors which can be ruled out promptly that are frequently blamed for this trouble. The first is lack of lubrication, unless the lubrication ceased entirely and the piston was completely destroyed. We have never seen a case in the average high-speed internal-combustion engine where lack of lubrication would cause breakage of the rings or ring lands, as the piston will scuff and score or the rings will freeze tight in the ring groove before breakage will occur. The blowby which occurs under certain conditions of insufficient lubrication will, however, cause breakage of the rings and ring lands by the excessive heat developed, causing the rings to stick in the ring grooves.

The first thing we must consider in attempting to analyze a difficulty of this kind is whether the ring broke, causing failure of the ring land or whether the ring land failed, causing breakage of the ring. If we are given an

opportunity to examine the piston shortly after the failure occurs, we can frequently determine which failure occurred first, but, unfortunately, if the piston has continued in operation for many miles following the first failure, any signs which would guide us have become obliterated, and from that point on we must assume so much that we can never be definitely sure of our conclusion.

Of course, as is frequently the case, when we look at this piston the ring or the ring land may not be broken and this automatically decides for us which part failed and caused the trouble. If both the ring land and ring are broken, there are some conditions that we can look for which will indicate the proper cause. First, where is the ring broken? In 99 cases out of 100, if a ring is broken at the heel or approximately 180 deg from the joint, the failure was caused by some defect in the piston itself, as this is the strongest part of the ring and a ring does not break at this point from ring flutter or other ring trouble. When a ring breaks in a ring groove, the small pieces of the ring work up and down in the groove and very often will hammer their way into the top or bottom of the groove, and in some cases we have seen where they actually have worked through the head of the piston. If ring failure occurred before the ring land failure, evidence of this working action of the small pieces can usually be found on several other parts of the groove, and the edges of the break in the land will be definitely radiused with the radius on the side of the land where the ring broke. If the ring land failed first, the land will be tapered away from the break, starting at the break where the land may be worn approximately halfway through, back for a distance of approximately 1 or 2 in. where the land assumes its normal width and, consequently, the ring groove is of normal width. This condition is brought about by the lack of support at one point in the land, allowing the ring to flex and causing wear on the ring land at each side of the break, moving progressively away from the break as operation continues. Of course, from this it can be seen that after all the ring lands have been broken off and the rings have to be picked up with a magnet to get all of them, there is not much indication left as to where failure first occurred.

■ Conditions behind Ring and Land Failure

Our next problem is to determine what condition was behind what we see actually to cause either the failure of the ring or the ring land. The one condition which is most frequently the cause of this breakage, and which may cause breakage of either the ring land or the ring itself, is *detonation*. It does not require much detonation to cause piston or ring breakage. For example, if a job runs out of water or the fan belt fails, detonation under the overheated condition occurs very quickly and the pistons can be ruined merely by driving the job a mile or two to the shop. The fact that the ring or piston failure was caused by detonation usually can only be determined by the elimination of the other factors which can cause ring-land or piston-ring breakage.

■ Improper Installation

The installation of piston rings with a lack of sufficient side clearance in the top groove will usually result in ring sticking and very often in ring breakage, particularly if the ring only sticks at one point. Under this condition, ring breakage most frequently occurs at the points adjacent to the point where the stickage occurred. The second factor causing ring breakage which must be looked for is the installation of piston rings in excessively worn ring grooves. Unfortunately, ring grooves do not wear evenly at all points and where wear is excessive a definitely wavy condition exists in the ring grooves, supporting the rings at various points but leaving the ring unsupported between these points. The stress of operation soon causes a fracture at one or more of the unsupported points. Unless the piston has been damaged very seriously by long operation following the failure of the ring land, it will usually be found that the ring groove on the piston at 180 deg from the breakage is approximately the groove width at the time the ring was installed, and the condition of the groove can be checked by inserting a new ring and a feeler gage in the undamaged portion of the groove.

■ Insufficient End Clearance

Another factor that will cause a breakage of piston rings is the installation of the piston ring with a lack of sufficient end clearance. Unfortunately, once a ring has been broken there is no way that this can be definitely determined, but this trouble is by no means as prevalent now as it was several years ago. The average maintenance man is conscious of end clearance and is not as afraid of blowby or oil consumption through the ring joints as he once was.

Laboratory tests which have been run on engines in normal operation have shown that blowby does not reach an excessive figure until end clearance exceeds 0.060 in., at which time the increase is probably caused more by ring flutter and poor ring shape than by actual leakage through the joints. Of course, the foregoing applies only to the average automotive-type engines that we have been discussing and not to the very slow-speed diesels and air compressors.

It is of course understood, in connection with ring breakage, that if care is not used during installation, they can be broken at this time, and this will not usually show up in the engine until it has been in operation for 4000-5000 or more miles.

The piston-ring lands can very readily be fractured by removing the piston from the cylinder without first removing the ridge at the top of the cylinder. When pistons are removed, if this ridge is quite severe, it is necessary to use considerable force to force the rings to jump over the ridge, and the impact of the rings striking the ridge frequently breaks or fractures the land immediately below the top ring. With a fracture of this kind, it will not be noticed when the piston is cleaned up unless it is very carefully inspected, and frequently inspection will not reveal the weakness unless the fracture has come all the way to the outside edge of the land. The forcing of pistons into a cylinder, using a defective ring clamp or one that is not properly tightened, may cause either breakage of the bottom oil rings or fracture of any of the lands on the piston. Likewise, if an engine has become badly worn, leaving a considerable ridge at the top of the cylinder, and a bearing fails or a rod bolt snaps, allowing the piston to travel further up in the cylinder than normal, causing the top ring to strike this ridge, breakage of the ring land between the first and second ring grooves may occur. After we have eliminated the various items mentioned above, we

can be almost definitely certain that the difficulty was caused by detonation.

If we will observe a few very simple precautions, we can completely prevent a recurrence of this difficulty in the future:

1. On all engines where it is possible, remove the ridge from the top of the cylinder before removing the piston, to prevent breakage of the ring land on the piston caused by striking this ridge.

2. Make certain that the ring grooves in the piston are not worn excessively and that the sides of the ring grooves are straight and free from uneven spots, nicks, burrs, or any other blemishes.

3. Use only rings manufactured by a reputable manufacturer of high-quality merchandise.

4. Fit the rings with a side clearance of 0.002-0.005 in. for the top groove, and 0.0015-0.0045 in. for the remaining grooves. This applies to 4-cycle gasoline engines. Four-cycle diesel engines require about 0.001 in. more. Two-cycle gasoline engines should be set up with side clearance of 0.004-0.008 in. in the top groove and 0.003-0.006 in. in the remaining grooves. Two-cycle diesel engines require still more clearance, and we would recommend on these 0.006-0.010 in. on the top groove and 0.004-0.008 in. on the remaining grooves. Use a minimum end clearance on all rings of 0.003 in. per in. of cylinder diameter in 4-cycle gasoline engines, and 0.004 in. per in. of cylinder diameter on other types.

5. Use care when installing pistons in the cylinder to be certain that the rings do not strike the top of the block when the piston is forced in and that the rings are not damaged when they are closed up in the ring clamp.

6. Make certain that the engines operate without destructive detonation.

In passing, we might point out that detonation can be destructive without being audible and the only way to make definitely certain that detonation does not occur is to adjust the engine properly for the particular grade of fuel which is being used and not to change fuels without changing these adjustments.

■ Excessive Sludge Formation

The second main problem that has been brought about by wartime operation of engines is that of the formation of excessive quantities of sludge in the crankcase. While this applies somewhat to fleet operation, it applies more particularly to the light delivery trucks and passenger-car engines that are operating on short runs at slow speeds with considerable idling. These conditions result in extremely low crankcase temperatures, accelerating sludge formation which eventually will clog up the oil lines, oil intake, and oil control rings on the pistons. As covered later in the report, the easiest preventive measure that can be adopted cannot in reality be called a preventive measure as it will not prevent this sludge formation completely, but it will at least reduce the formation and make it somewhat easier to control, and that is the use of a somewhat lighter viscosity oil to get a more rapid circulation of oil and therefore a better flushing action, and the use of some type of detergent oil where it is available. Particularly in passenger cars, this condition indicates definitely that oil changes should be made at more frequent intervals.

The poorer grades of gasoline, together with this lower operating temperature, increases crankcase dilution considerably and the formation of acids in the lubricating oil seems to be quite rapid. A word of caution, in connection with this, concerning oil filters, for if a filter is used that has a fine wire screen or retainer for the filter element, difficulty will probably be experienced by the attack of the acid on the wire retainer causing its breakdown, under which condition the lubricating oil carries small particles of the screen through the oiling system into the bearings. When this happens, usually a complete new set of bearings as well as a new or reground crankshaft, and possibly camshaft, is required.

Operating an engine with excessive crankcase dilution not only causes difficulty from the acids, but destroys, to some extent, the lubricating value of the oil, which results in rapid wear and in some severe cases in scuffing or scoring.

■ Rapid Formation of Varnish Deposits

The third main problem which is being encountered during these times is the seemingly more rapid formation of lacquer and varnish deposits throughout the engine, and particularly on the pistons and cylinder walls. We have known of cases where vehicles which have been laid up for several months could not be operated because of the large deposits of varnish and lacquer in the cylinder walls. After about five miles of operation this varnish and lacquer would soften up to a point where if the engine were stopped it could not be started again, as the adhesion of the pistons to this varnish formation was so great that the engine could not be turned over. This of course is an extreme case, but this formation of lacquer and varnish usually shows itself following the major overhaul and reringing of an engine in the form of excessive oil consumption for often as long as 3000 or 4000 miles.

■ Glazed Cylinder Walls

This leads us into another channel of trouble, and that is glazed cylinder walls, as the lacquer and varnish formation actually creates the same conditions in the cylinder as does an excessively glazed wall. For the operation of an engine, it is quite true that we want a smooth cylinder wall and one that is properly rebored or honed, but we do not want one that is glazed by the use of dirty stones in the hone or any of the numerous factors that can cause this condition. In the engine that is to be reringed without reboring, the particles of varnish and lacquer fill the pores in the metal of the cylinder and create the same condition that has just been described. Under these conditions, lubricating oil does not adhere to the surface of the cylinder and it is not possible for the rings to distribute the oil satisfactorily through the length of the cylinder. What oil is left on the cylinder rides ahead of the rings instead of forming a film between the ring and the cylinder wall, resulting in excessive oil consumption and in many cases excessive ring wear, depending upon the actual conditions in the engines and the amount of glaze. Instead of resulting in excessive ring wear, it may result in no ring wear at all, preventing the rings from seating themselves and forming the proper contact with the cylinder walls.

The condition of excessively glazed cylinders is one that is very easily overcome, and in fact many of the better garages and fleet operators have made it a practice to use one of the emery cloth devices, such as our glaze buster or

spring-type hone, which are both designed especially to move the glaze without removing any of the cylinder aterial.

Any discussion of piston, ring, or cylinder problems ould not be complete without including a discussion of owby and scuffing or scoring, and as they are encountered quite frequently by those interested in all types of ternal-combustion engines, I believe that they have a efinite part in this discussion, even though they have not en affected to any great extent by present-day operating nditions. Under the definition of a piston ring, we ated that one of the duties that a piston ring was expected perform was to prevent excessive quantities of combuson gases from passing between the piston and the cylinder alls. At first glance this appears to be quite a simple job, ut, unfortunately, there are a number of factors which nter into the picture and which make this job considerably ore complicated.

First of all, we must assume that a high grade of ring as been properly installed on the piston and the piston arefully placed in the cylinder. As we look at the cylinder, o use the ideal condition, that has been rebored, we are clined to think of this cylinder as a perfectly round hole n which the piston travels up and down, but as soon as e begin to attach other parts to the engine we begin ulling these cylinders out of shape. This applies to large ngines as well as small engines and I know of no engine hich is not affected in this way. As the head is pulled own, the cylinder is distorted even if it is pulled down bsolutely uniformly, but if the same pressure is not used n each stud the condition is immeasurably worse. As we ighten up the main bearing caps, the manifold, and so orth, we tend to distort the cylinder even more, so that y the time we have completely assembled the engine and t is ready to be started we have anything but a round traight cylinder in which the rings and pistons are to unction. This distortion is serious enough if all of the olts are pulled down uniformly to the manufacturer's ecommended tension by using a tension or torque wrench, ut if some mechanic throws a wrench with a piece of ipe on it on one stud and pulls down particularly hard ecause he thinks this one stud may be loose, he can very eadily destroy any chance that this engine has to give us atisfactory performance.

Use of Torque Wrench

By all means, a tension or torque wrench should be used throughout when assembling an engine, whether it has been rebored or simply reringed. Those who do their own reboring will find that they will get better jobs if they tighten the main bearing caps up to the specified tension, with or without the shaft in the engine, and install a dummy head pulled down to the specified tension before they begin their reboring operation. This practice gives us the closest approach that we can get to a round straight cylinder.

We now find that we have a cylinder that is anything but round, but if we have taken all the precautions possible, there is not much that we can do about it, and piston rings are designed to take care of the distortion, provided it has been kept to a minimum.

We now start the engine, and immediately the cylinder walls begin to heat up. If we have a cooling water jacket which has collected deposits in the bottom or on the outside of the cylinder, there is no cooling at the points where this deposit occurs and, therefore, this one section of the cylinder operates at a higher temperature than the remainder and we have a pocket due to the uneven expansion. If the average engine is rebored to a small oversize there is ample material left in the cylinder wall to provide adequate cooling, assuming that there are no deposits in the cooling system, but if we take an engine and rebore it to a large oversize, any slight differences in the thickness of the cylinder walls become more sensitive to temperatures and again we have uneven cooling, resulting in pockets. Under these conditions, no piston ring can give satisfactory blowby control in 100 out of every 100 engines overhauled. To the mechanic who is working on an engine that has dry sleeves installed in it there is one additional caution, and that is that no deposits must be present between the sleeve and the block. These deposits act as insulators, causing hot spots.

Effect of Hot Spots

We had one case of a large Midwestern operator who had a fleet of trucks equipped with dry sleeves and after approximately every 70,000 miles the blowby would become so serious that the fumes from the breather would make it almost impossible for the driver to stay in the cab. An investigation revealed that this engine was so designed that the cylinder-head gasket did not cover the top of the joint between the sleeve and the block and carbon was forming between the sleeve and the block. These sleeves were pulled from the block and a wire brush was used to remove the carbon formation and the bore in the block was cleaned up with a hone, at which time the sleeves were again placed in the engine and no blowby was experienced for another 60,000-70,000 miles of operation. This indicates just what effect hot spots have on blowby, and this operator has now made it a practice to pull these sleeves and clean them up whenever they perform a major overhaul. Since that time he has experienced no further difficulty with blowby.

It is quite possible to cause serious distortion of sleeves, particularly of the very thin type, during installation, and, by all means, the sleeve manufacturer's recommendation for installation should be followed absolutely to the letter. While it is rather bothersome, the most satisfactory means of fitting these very thin sleeves is by cooling them first with dry ice and then inserting them in the cylinders, allowing them to expand after they are in the bore. Efforts to force them in by means of clamps or any other devices frequently result in partially crushing the sleeve, causing waves in them and poor contact with the bore.

Blowby can also result indirectly from the excessive formation of sludge in the crankcase, as it is necessary that the compression and oil control rings on a piston function in conjunction with each other. If an oil control ring becomes filled with deposits so that oil control is no longer adequate, the large quantities of lubricating oil reaching the compression rings soon cause them to stick in the ring groove, resulting in excessive blowby. Likewise, the excessive blowby caused by cylinder distortion allows the hot gases from the combustion chamber to reach the oil control rings, solidifying the deposits which are constantly present at this point, and burning the oil, causing additional deposits which rapidly fill up the oil control rings. Then we have an engine on our hands that must be given a major overhaul.

Breakdown in Lubrication

There have been many articles and books written on the subject of piston scuffing and scoring, but for our purpose we can reach one definite conclusion which will answer our purpose, and that is, regardless of where it occurs, scuffing, scoring, or excessive wear is brought about by one condition, and one condition only, and that is, a breakdown of the film of lubricating oil between the piston and its rings and the cylinder wall. Our problem in connection with scuffing or scoring is to determine what caused this breakdown in lubrication and, of course, excessive overheating of the engine or failure of the lubricating system are obvious causes. However, there are other conditions which will cause a breakdown in this lubricating film and we will consider them, not necessarily in the order of their importance.

Excessive blowby can burn or remove the film of lubricating oil from the cylinder walls, resulting in metal-to-metal contact and scuffing or scoring.

Excessive quantities of raw gasoline in the combustion chamber wash the lubricating oil from the cylinder wall, allowing metal-to-metal contact and scuffing or scoring. In the passenger-car and light-delivery fields, automatic chokes are the most common causes of this excessive gasoline, but sticky floats or generally defective carburetors can likewise give trouble.

Improper clearance or grinding of the piston can cause sufficient direct pressure when the piston expands to cause a breakdown in the lubricating film and the scuffing and scoring. In this connection, those who grind their own pistons should, by all means, follow the exact recommendations when grinding pistons that are set up by either the manufacturer of the vehicle or the manufacturer of the piston. We can allow what appears to be adequate clearance on the skirt of the piston, but if we do not allow enough cam for the piston to breathe under operating temperatures, the skirt of the piston will force itself against the cylinder wall with such pressure that scuffing and scoring will definitely result.

Fitting of Piston Pins

Along this same line, particularly in the lighter-weight, more flexible pistons, improper fitting of piston pins will cause the pistons to score on the sides in the general area of the pin bosses or immediately below them. When we fit pins, we must allow sufficient clearance, at least on one side of the piston, to allow for movement between the pin and pin boss so that when the pin expands it will not force the skirt of the piston out against the cylinder wall, or when the piston expands it is not restricted at the pin bosses, forcing it to expand at 90 deg from the pin.

The improper installation of piston rings can likewise cause scuffing and scoring by too severe oil control, brought about by a desire on the part of the operator or the supplier of the rings to decrease oil consumption blindly to a point where adequate lubrication is not provided.

Determining the Cause of Scuffing

When faced with a condition of scuffing or scoring, our problem is to determine which one of the above factors was responsible for the trouble, and as we hold the piston with its rings in our hands there are certain things to look for which will indicate where our trouble originated. Our first problem is to determine on what part of the piston or rings the scuffing or scoring originated.

While not invariably true, experience has shown that the point where the scuffing or scoring originated is usually the point where the maximum scuffing or scoring appears. Scuffing, unless it completely covers the piston, is usually roughly in the form of a triangle or ellipse with the more dense section in the center of the ellipse or at the base of the triangle. The lines leading out from the point of greatest density are where the scuffing has progressed, caused by the roughing up of either the piston or cylinder, as the case may be. If we find that the greatest amount of scuffing or scoring appears to be in the ring belt of the piston becoming less dense as it continues down the skirt of the piston, we can assume that the trouble was caused by a lack of lubrication in the ring belt area caused by blowby or too severe installation. While on this subject, the top lands of the piston tend to grow in operation and before a piston is ever reinstalled in an engine the diameter of this top ring land should be checked carefully to make sure that it does not exceed the size recommended by the vehicle manufacturer or the piston manufacturer. In the absence of such information, the top land should be at least 0.030 in. smaller in diameter than the nominal size of the cylinder.

If we find that the scoring on the piston has originated on the skirt, we can then reasonably determine the cause of the scoring by its location. If it originated in the area of the pin bosses or immediately below them, we are reasonably sure that it was caused by improper fitting of the pins or a lack of sufficient cam on the piston. If it is on the thrust or counterthrust side of the skirt, we can assume that it was caused either by a lack of sufficient clearance when the piston was installed, or a breakdown in the lubrication system, or overheating. Scuffing or scoring, which appears indiscriminately around the greater part of the skirt of the piston, indicates too little cam on the piston and possibly too little clearance.

Of course, all of the above conclusions are reached on the assumption that it has been determined there was no excessive overheating of the engine or breakdown of the lubricating system.

Method of Prevention

The method of prevention in these cases is obvious: Use proper clearance and cam when installing pistons. Fit the pins properly and maintain the lubricating system in good condition.

In the early days of the flexible type of ring it is quite true that, just as with any other product of a similar nature, there was considerable difficulty experienced, and for this reason the flexible type of ring has got a bad name with a great many fleet operators and mechanics in general. The quality and performance of the flexible type of ring has greatly improved, and can now be used in almost any type of operation to conserve labor, cylinder blocks, and other parts, by making it unnecessary to rebore blocks when the engine is reconditioned. Many fleets have adopted the use of a set of flexible rings between each rebore job, increasing by approximately 50% the life of the cylinder blocks, which saving, at the same time, is very necessary and desirable because of the scarcity of parts. If those who have experienced difficulty with the flexible type

of rings and are not now using these rings, will give them another trial, we know that it will result in a saving in man hours and material.

■ Avoiding Future Troubles

Now that we have discussed some of the problems and have seen their effect on engines, what can we do to avoid these troubles in the future and secure the maximum mileage from each major overhaul at the lowest possible cost and with the greatest efficiency? The first caution that will overcome a tremendous amount of our difficulty is, above all, *avoid detonation.* This will eliminate many of the more serious problems that we have, and at the very least will reduce our problem in connection with replacement parts, particularly pistons, rings, and bearings.

This leads us to the second precaution, which actually should be placed first, and that is *proper maintenance*. Proper maintenance will prevent destructive detonation. It is true that most shops are greatly overloaded, but if two hours of checking an engine result in prolonging the efficient life of an engine between major overhauls by only 5000 miles, it is time well spent, as it will reduce in the long run, particularly with fleets, the number of major overhauls requiring in some cases several days and resulting in a total waste in manpower-hours when they are so precious. I do not believe it is necessary to mention that with the manpower situation as it is, it is false economy to use anything but the best replacement parts available. Early in the emergency there seemed to be some individuals who attempted to place trick gadgets and inferior materials on the market for use by the gullible mechanic but, fortunately, the government seems to have done, in the main, a pretty good job of restricting this practice.

Whenever it is necessary or desirable to use worn parts over again in an engine, as a safeguard, one should be certain that these parts are in usable condition. The installation of a cracked piston, for example, will very probably result in failure of the piston and possibly complete destruction of the engine. If the ring grooves are worn somewhat when the piston is installed in the engine, ring breakage and destruction in the piston will probably result; but, if on the other hand, sufficient time is taken to have these pistons regrooved to the next standard width ring there is every reason to believe that it may be used satisfactorily for many more thousands of miles.

One caution, particularly when one encounters an engine which may be a little older than usual or an engine that is not a very popular model: if the engine must be placed back on the road immediately, one should be certain that all of the parts which it is likely will have to be replaced are available before the engine is torn down. Proper fitting of pistons, rings and sleeves, while it may take a little longer during the overhaul job, may result in double the amount of mileage before the next overhaul is necessary and, needless to say, if this can be accomplished, it is well to take a little extra time to be certain that all parts are fitted according to the specific instructions of the manufacturer of the parts or of the engine.

Particularly in cases where engines are used on short runs at slow speeds and engine crankcase temperatures remain very low as a result, it may be wise, according to the particular type of operation, to use a lighter grade of lubricating oil than has been used in the past, and the use of a detergent type of oil, if this type is available, should be definitely considered, as the lighter oils and detergent oils will tend to prevent the formation of heavy carbon and sludge deposits in the rings and oiling system of the engine by the flushing action of lighter oils and the solvent action of the detergent oils.

Numerous times delivery trucks particularly stand loading or unloading with their engines idling for long periods. This is a problem principally for the fleet operator. I recently had occasion to notice a large tractor standing at the loading platform at our plant. I was interested to note that this engine idled for 1 hr and 20 min while the vehicle was being unloaded. This is all unnecessary engine mileage which does not appear on your speedometer or odometer but which is nevertheless causing the engine to operate for unnecessary periods of time under the worst operating conditions possible. Under idling conditions, combustion of the fuel in the cylinders is often incomplete and a tremendous amount of crankcase dilution results. The lubrication system of the average engine of today is designed for maximum efficiency at operating speeds, and proper lubrication is extremely doubtful under idling conditions when crankshaft speeds are low, particularly when extended over a protracted period of time. The only cure for this is to impress upon the car owner, in the case of passenger cars, or the truck operator in the case of the fleet operator, that idling time should be reduced to an absolute minimum to save the engine, even if he is not sufficiently patriotic to avoid this unnecessary waste of fuel.

■ New Developments

It has been said many times that the scientific developments of any nation go ahead 20 years in any one year of war, and the automotive industry, as reflected by the developments which have been made in connection with piston rings, certainly bears out this old saying. There have been new materials developed, new processes developed, and new theories evolved and proved which, when the present emergency is over, will make all of our jobs very much easier. The possibility of piston rings which will give satisfactory performance for 350,000 to 400,000 miles under proper maintenance conditions is definitely within the realm of possibility. It has already been done on tests and there is no reason to believe that there is anything that will prevent these rings from giving equally satisfactory performance both in the passenger-car field and the truck field, when they become available.

■ Future Considerations

New theories concerning lubrication systems have been developed, and everything seems to point to engines which instead of being overhauled every 50,000 to 75,000 miles will probably last, as mentioned above, for 350,000 to 400,000 miles. Porous chrome, while considerably different from the chromium we see on many articles in common use today, is nevertheless of the same family, and seems to have a definite place in the post-war picture. Cast-iron rings which will not break under any conditions, such as detonation, which we have covered before, are a probability, and coupled with developments now in the process of perfection will make life, as far as pistons, rings, and cylinders are concerned, very much more pleasant.

COMPARISON of LABORATOR[...] with SERVIC[...]

by R. S. WETMILL[...]nd

GENERAL MOTORS two-cycle model 71 diesel engines have been used for the past several years to evaluate the performance of lubricating oils by the 500-hr laboratory test procedure. In the course of development work on heavy-duty lubricants, this test indicated two oils to be particularly outstanding, and it was considered desirable to accumulate service data on their performance. At the same time, it seemed important to have available service information on other less desirable oils in order to interpret results of laboratory tests properly.

To these ends, a field test of approximately 580,000 engine-miles was carried out on GM diesel powered buses through the cooperation of the Greyhound Corp. Results so obtained were compared with those previously determined in 500-hr laboratory tests on the same oils.

[This paper was presented at the SAE Diesel Engine and Fuels & Lubricants Meeting, Cleveland, Ohio, June 2, 1943.]

■ Testing Procedure and Results

Five high VI additive-type oils of SAE 30 grade were used in both laboratory and field tests. Diesel fuel of 51 cetane was used throughout.

Laboratory runs were conducted in three GM 3-71 diesels in accordance with the GM 500-hr procedure. Operating conditions used are shown in brief in Table 1. Field tests were conducted in 10 GM 6-71 diesel powered PDG-3701 buses operating their regularly scheduled runs from Detroit, Mich., to Chicago, Ill., and Columbus, Ohio; two buses were operated on each oil. Close supervision was maintained at all times by personnel permanently assigned to the project. Servicing and inspections were made at approximately 550-mile intervals as shown in Table 2. Runs were continued to approximately 60,000 miles unless failure necessitated premature shutdown.

Results of laboratory tests are compared with those ob-

THE General Motors model 71 diesel, 500-hr laboratory test procedure yields results that are indicative of the service performance of lubricating oils, according to the authors. However, they caution, the oil performance is depreciated somewhat beyond that encountered in reasonably heavy-duty service operations due to the severity of the 500-hr test.

The 500-hr laboratory tests and the 60,000-mile service tests are compared on the basis of bearing corrosion, ring sticking, filter clogging, engine deposits, wear, and used oil contamination.

The oils were rated in the same order by both the laboratory and the service tests. Agreement between the two tests was very good with respect to bearing corrosion. Corrosion results of both tests were also found to compare favorably with data obtained in the MacCoull corrosion tester. Similarly, good agreement between the two tests was found with respect to ring sticking, oil ring deposits, piston deposit, Purolator clogging, and used oil examination.

Differences in air port deposits were apparent in the laboratory tests, but in the field these deposits were so light that no differences could be detected. In a similar manner, differences in AC filter clogging were experienced in the field, while in the laboratory, where the filter was changed whenever the oil became dirty, no differences were found.

Cylinder liner ridging was found to be of a random nature, the only consistent observation being that liners with low mileage exhibited little or no ridging. No differences in ring wear were observed between the several oils; however, it appears that wear does not become appreciable until the liner has operated approximately 100,000 miles.

★ ★ ★

THE AUTHORS: R. S. WETMILLER has been assistant supervisor of engineering research with The Texas Co. since 1940. Before engaging in fuels and lubricants work for that company in 1937, Mr. Wetmiller spent two years as research assistant at Pennsylvania State College, where he had received his M. S. degree in mechanical engineering. BRUCE HEGEMAN who, shortly after his graduation from the U. S. Naval Academy in 1927 joined The Texas Co., is now supervisor of product applications at its research laboratories in Beacon, N.Y.

IESEL-ENGINE TESTS
ERFORMANCE

d BRUCE HEGEMAN *The Texas Co.*

tained in field tests with respect to bearing corrosion, ring sticking, and pertinent engine deposits in Table 3. More detailed tables showing results of individual runs are included as Appendices I to VIII. These tables also serve to show that satisfactory reproducibility was obtained, at least in field tests where duplicate runs were made.

Only two of the samples tested caused any appreciable copper-lead bearing corrosion, oils A and B. Laboratory and field results agree exceptionally well, as shown in Table 3 and by the photomicrographs in Fig. 1. A comparison of bearing weight losses obtained in the two tests is given in Fig. 2; good agreement is again indicated. It is interesting to note that the slope of the curve is approximately ¼; this is probably due to the fact that service tests were run for 60,000 miles in 6-cyl engines and laboratory tests for 30,000 miles in 3-cyl engines.

Bearing corrosion performance was also compared with

Table 1 – Operating Conditions – 500-Hr GM Diesel Test

Engine	GM 3-71
Speed, rpm	2000
Load, bhp	80 ± 2
Water Out Temperature, F	180 ± 3
Intake Air Temperature, F	105 ± 5
Fuel Manifold Temperature, F	120 ± 5
Oil Temperature, F	230 ± 1
Exhaust Back Pressure, in. Hg	8
Oil Consumption, lb per hr, maximum	0.21
AC Filter Flow Rate, gph	30

Engine trichlorethylene degreased and assembled with new pistons, rings, liners weighed Cu-Pb bearings, cam followers, and such other parts as required to maintain manufacturer's recommended clearances.

Regular test preceded by 16-hr break-in and 5-hr adjustment run.

AC filter cartridge installed at 144 hr; cartridge replaced as required when oil became black.

Purolator strainer replaced when pressure drop across it exceeded 10 psi.

Air ports cleaned when air box pressure rise exceeded 3 in. Hg.

Engine inspected and one main and one rod bearing replaced at 250 hr.

Table 2 – Servicing and Inspection Schedule – Field Tests

550-Mile Intervals—Check oil, general maintenance servicing.

5000-Mile Intervals—Photograph, clean, and reinstall Purolator strainer. Photograph and reinstall AC filter cartridge. Obtain sample of crankcase oil for analysis.

10,000-Mile Intervals—Photograph, clean, and reinstall Purolator strainer. Photograph and renew AC filter cartridge. Obtain sample of crankcase oil for analysis. Change crankcase oil. Examine air box, intake ports, pistons, and rings through air box cover plates.

60,000 Miles—Test termination—dismantle, inspect, and photograph engine parts.

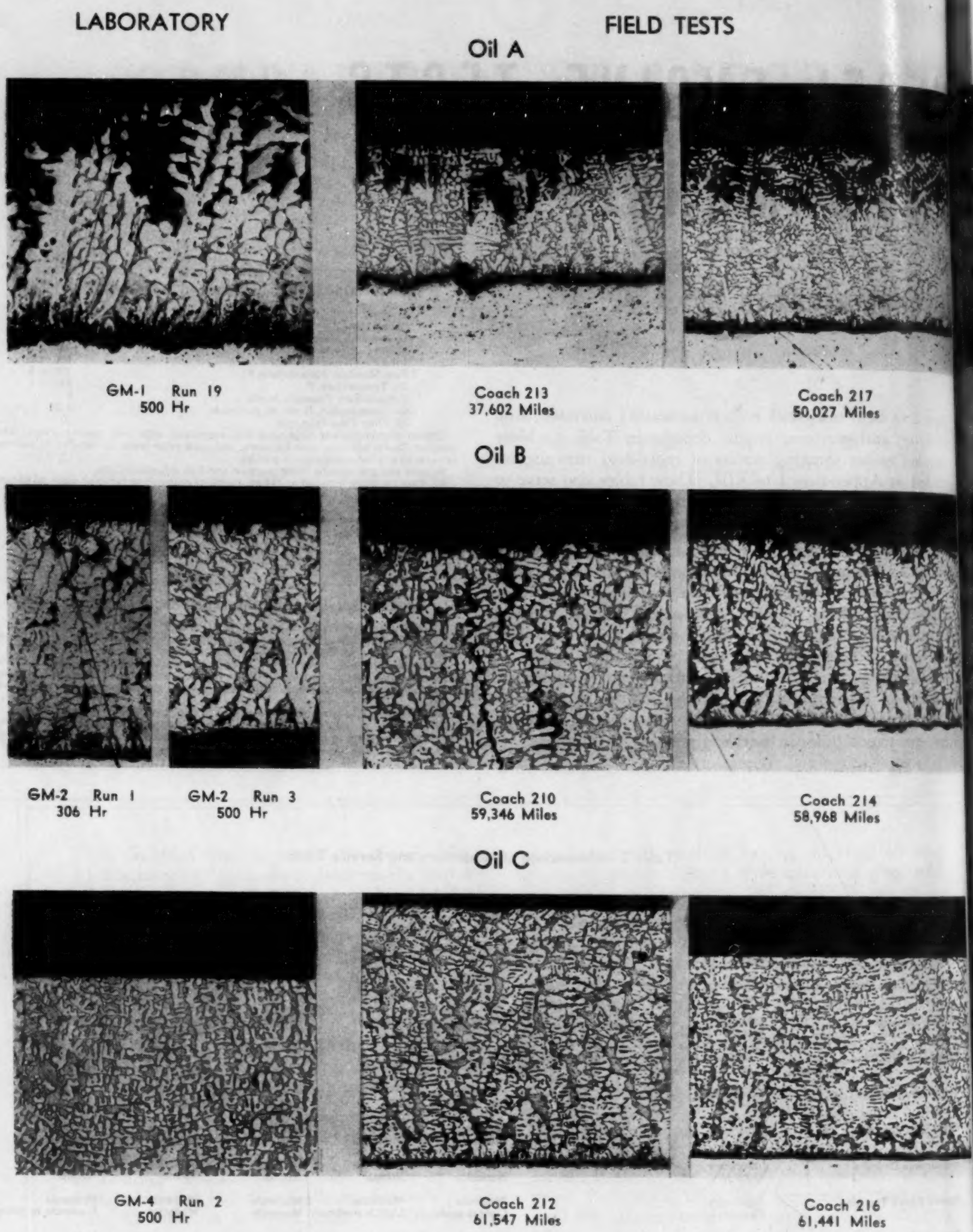
Engine boiled in Oakite and assembled with new weighed rings and Cu-Pb bearings and such other parts as required to maintain manufacturer's recommended clearances. In most cases it was necessary to turn oil rings to the scraping position during tests.

Table 3 – Comparison of Laboratory and Service Tests

Phase of Performance	Test Location	Oil Designation				
		A	B	C	D	E
Bearing Corrosion	Laboratory	Severe	Moderate	None	None	None
	Field	Severe	Moderate	None	None	None
Number of Rings Stuck per Engine	Laboratory	2	3	3	0	0
	Field	4	9	8	2	2
Oil Ring Deposits	Laboratory	Heavy	Moderate	Moderate	Light	Light
	Field (% Closure)	50	60	65	0	45[1]
Airport Deposits	Laboratory (Number of cleanings required)	1	2	0	0	0
	Field (% Closure)	0	15	10	25	15
Ring Wear, g per cyl	Laboratory			No data available		
	Field	0.96	3.02	3.84	2.41	....
Piston Skirt Deposits	Laboratory	Moderate	Heavy	Heavy	Moderate	Moderate
	Field	Moderate	Moderate	Moderate	Moderate	Moderate
Ring Groove Deposits	Laboratory	Heavy	Moderate	Moderate	Moderate	Moderate
	Field	Normal	Normal	Normal	Normal	Normal
Deposit Above Top Ring	Laboratory	Moderate	Moderate	Moderate	Moderate	Moderate
	Field	Light to moderate	Light to moderate	Moderate	Moderate	Moderate to heavy
Purolator Deposits	Laboratory (Number of changes required)	2	1	3	0	1
	Field	Trace to light	Light	Moderate	Trace	Trace
AC Filter Deposits	Laboratory (Number of changes required)	2	2	2	2	2
	Field	Clean to light	Heavy	Heavy	Clean to light	Light to moderate
Used Oil Examination	Laboratory			No appreciable oxidation		
	Field			No appreciable oxidation		

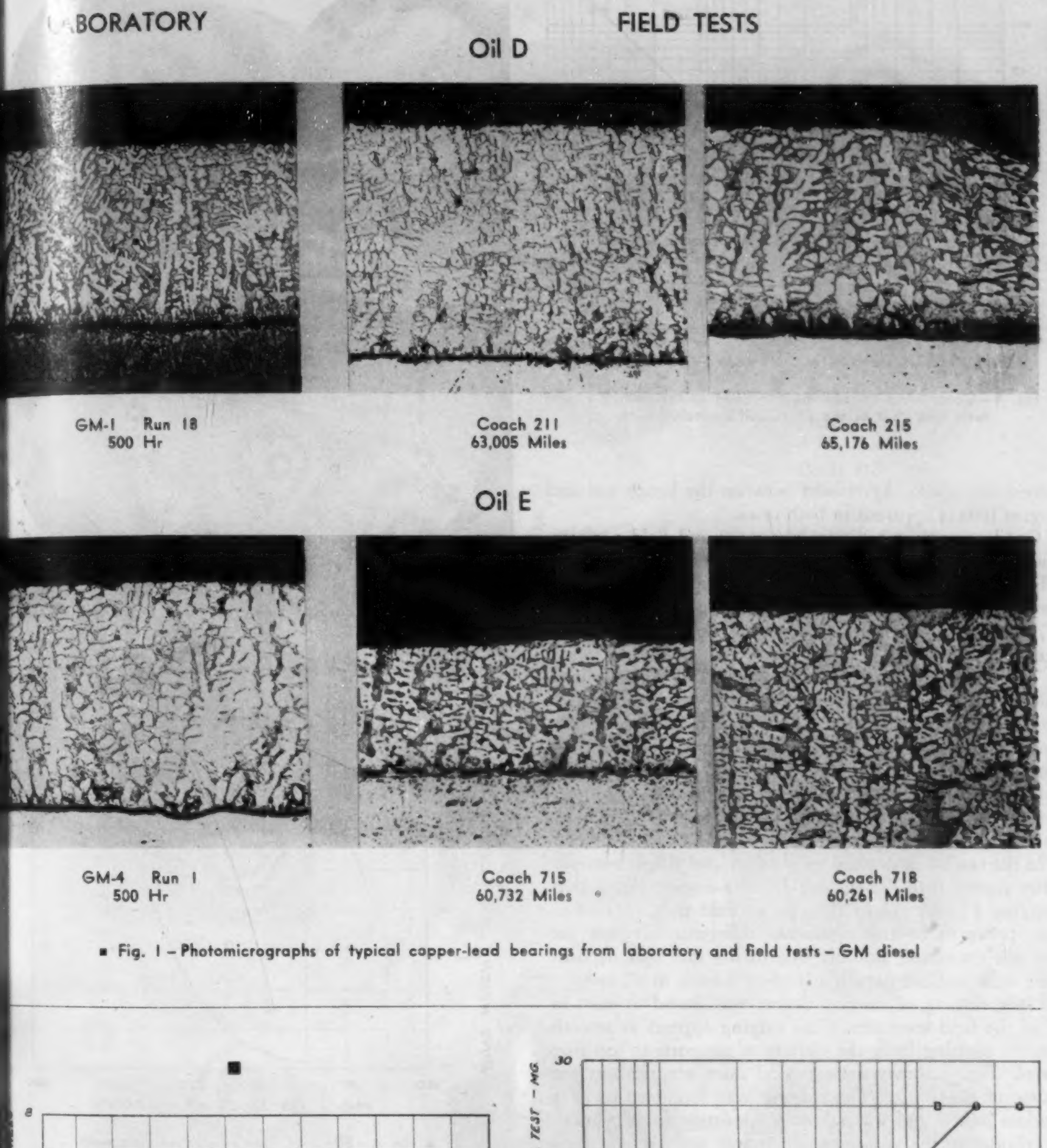
[1] Many compression rings found broken, heavy oil ring deposits probably attributable to high blowby rate.

Note: Field tests on oil A terminated prior to 60,000 miles because of bearing failure.

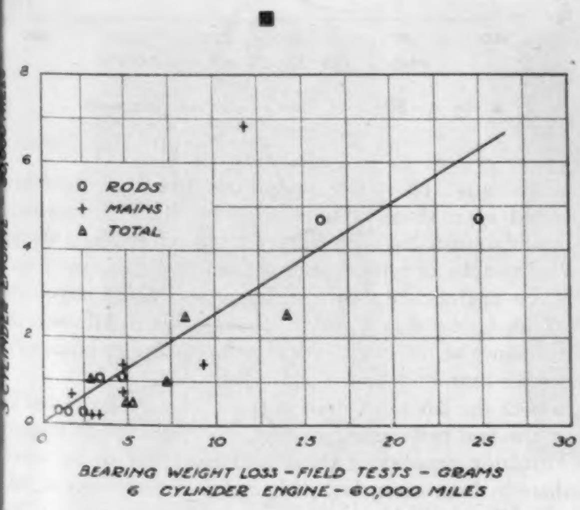


results obtained by the MacCoull corrosion test at 350 F.[1] In analyzing these data, MacCoull test bearing weight losses at 2-hr intervals from 2 to 10 hr were considered. The best agreement with engine results was found at 6 hr, as was the case with correlations on gasoline engines.[1] Fig. 3 is a plot showing the agreement with field performance and Fig. 4 with laboratory performance. The latter plot contains data on some 15 oils in addition to the five

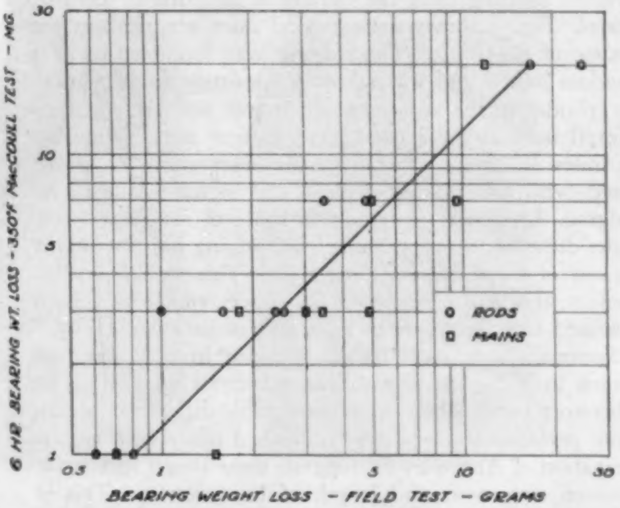
[1] See SAE Transactions, Vol. 50, August, 1942, pp. 338-345: "An Oil Corrosion Tester," by Neil MacCoull, E. A. Ryder, and A. C. Scholp.



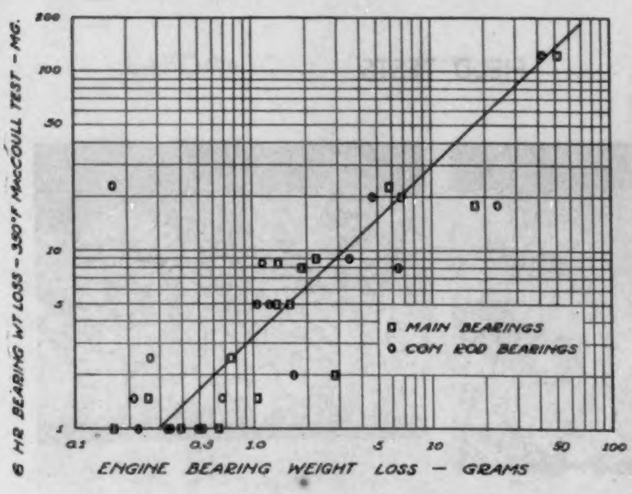
■ Fig. 1 – Photomicrographs of typical copper-lead bearings from laboratory and field tests – GM diesel



■ Fig. 2 – Comparison of laboratory and field test bearing weight losses



■ Fig. 3 – Comparison of bearing corrosion in field tests with that in the MacCoull corrosion tester



■ Fig. 4 – Comparison of bearing corrosion in laboratory engine tests with that in the MacCoull corrosion tester

herein discussed. Agreement between the bench test and engine tests is apparent in both cases.

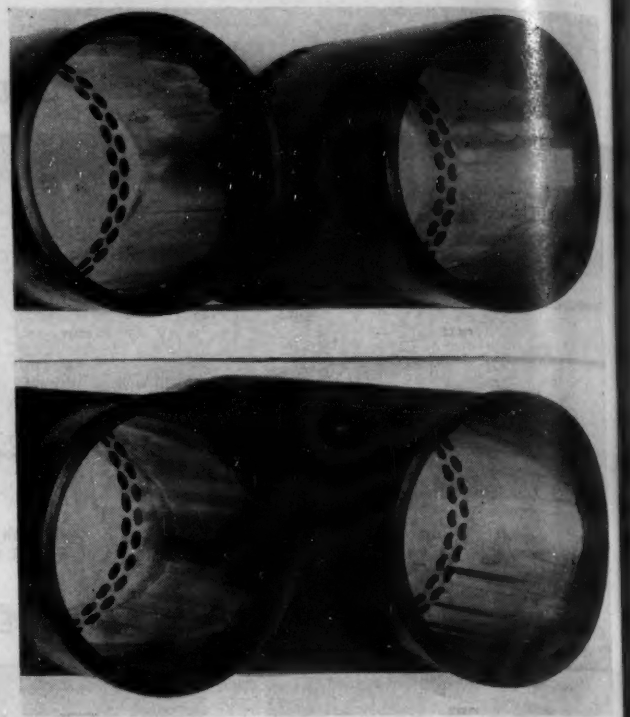
Good agreement between laboratory and field performance is observed with respect to ring sticking (Table 3). The oils can be divided into two groups; in both cases samples D and E show reasonably good performance while A, B, and C are much poorer and of about the same quality.

The oil ring deposit data summarized in Table 3 indicate good agreement between the two tests; samples D and E are superior to A, B, and C in both cases. The poor performance of sample E in field tests is discounted since both engines contained many broken compression rings (apparently due to mechanical causes), which would adversely affect oil ring deposits due to high blowby rates and resultant soot deposition.

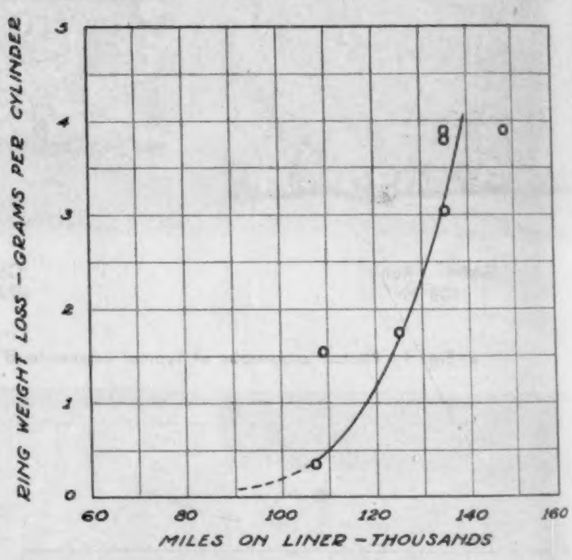
In the case of laboratory tests, oils A and B appear somewhat poorer than C, D, and E with respect to air port clogging (Table 3). In the case of field tests, there does not appear to be any significant difference between the five oils examined, possibly due to the fact that deposits were light and not particularly objectionable in all cases.

Liner ridging of varying degree was found to exist in all of the field test units. This ridging appears as smooth grooves running from the vicinity of air ports to top ring travel. Fig. 5 shows some typical cases observed in the course of the work. The ridging was found to be of a random nature and was not even consistent from cylinder to cylinder in the same engine. It was not influenced by the oil used and did not appear to bear any relationship to miles of service other than the observation that those liners with low mileage were found to have little or no ridging. In laboratory tests, which are of roughly 30,000-miles duration, no appreciable liner ridging has been noted on any of the numerous runs made. This would seem to confirm the field observation that liners which have been operated only short periods have little or no ridging.

Summary ring wear results obtained in field tests are shown in Table 3; no wear measurements were made in laboratory runs. There is no measurable difference in ring wear performance between the several oils; however, examination of Appendix IV suggests some rough agreement between ring wear and length of liner service. This is shown graphically in Fig. 6. Ring wear does not appear to be of any appreciable magnitude until the liner has operated approximately 100,000 miles. This performance is possibly associated with liner ridging as explained above.



■ Fig. 5 – Typical cases of liner ridging observed in field tests



■ Fig. 6 – Effect of liner service on ring wear

Photographs of typical pistons from both laboratory and field test engines are shown in Fig. 7 and visual inspection notes are included in Table 3. No significant difference in performance of the five oils is observed in either laboratory or service tests.

In both the laboratory and in the field, samples D and E show the best performance, and C the poorest with respect to Purolator deposits. Oils A and B appear to be intermediate in both cases, but their order of preference is not clearly defined (Table 3).

LABORATORY FIELD TESTS

Oil A

GM-1 Run 19
500 Hr

Coach 213
37,602 Miles

Coach 217
50,027 Miles

Oil B

GM-2 Run 1
306 Hr

Coach 210
59,346 Miles

Coach 214
58,968 Miles

Oil C

GM-4 Run 2
500 Hr

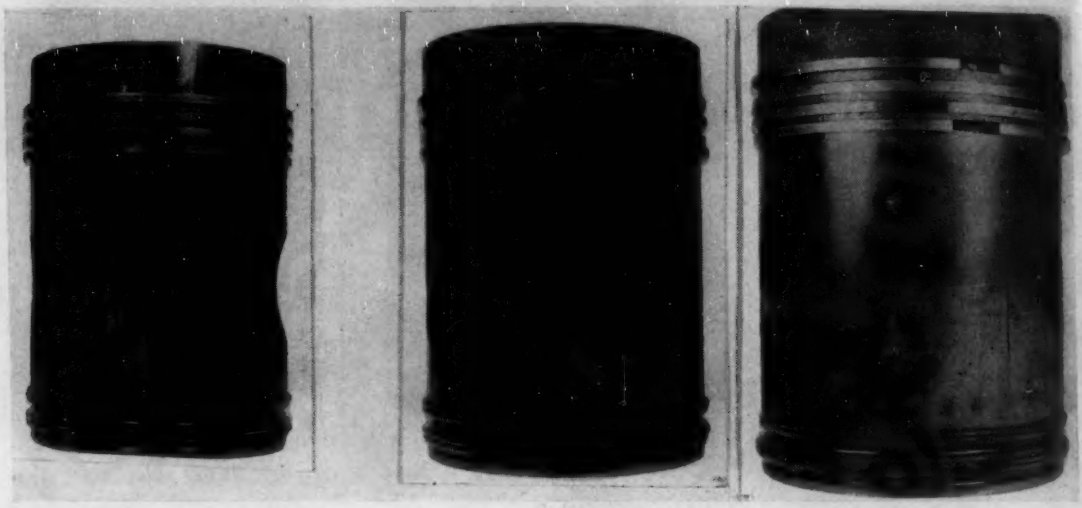
Coach 212
61,547 Miles

Coach 216
61,441 Miles

Cont.

LABORATORY   FIELD TESTS

Oil D



GM-1 Run 18
500 Hr

Coach 211
63,005 Miles

Coach 215
65,176 Miles

Oil E

No Photograph Available

GM-4 Run 1
500 Hr

Coach 715
60,732 Miles

Coach 718
60,261 Miles

■ Fig. 7 – Photographs of typical pistons from laboratory and field tests – GM diesel

No difference in AC filter deposits is observed between the five oils in laboratory runs, but in field tests important differences appear to exist. Oils A, D, and E are approximately equal and are superior to samples B and C (Table 3). In the laboratory procedure followed, the AC filter is changed when the oil becomes dirty rather than when the filter becomes dirty. This probably accounts for the fact that no difference between oils was observed in the laboratory tests.

Examination of used oils from both tests discloses no differences between the five samples in this respect. Oxidation was negligible in all cases.

■ Overall Comparisons

The relative performance of the five oils investigated is shown in Table 4. From inspection of this table, it is apparent that samples D and E are superior, and of about the same magnitude. Samples A, B, and C show much poorer performance with the exception of air port and AC filter deposits. In the case of field tests, air port deposits were not very objectionable in any case, thus, differences between samples could not be detected. In the case of laboratory tests, the AC filter is changed when the oil becomes black rather than when the filter becomes dirty;

hus any differences between filter clogging tendencies of the oils are minimized.

It appears that tests under the GM 500-hr procedure are, in general, more severe than field tests conducted under the methods used. Although laboratory tests were of only half the duration of field tests, they showed generally heavier deposits, particularly on the piston and in the air ports. This is probably associated with higher temperatures occasioned by the sustained full-load operating conditions employed.

Conclusions

The General Motors model 71, 500-hr laboratory test procedure yields results which are indicative of the service performance of lubricating oils. However, due to the severity of this test, their performance is depreciated somewhat beyond that encountered in reasonably heavy-duty service operations.

Acknowledgment

The authors acknowledge the cooperation of the Greyhound Corp. in making these field tests possible. The assistance and interest of W. A. Duvall and P. R. Ryberg and their assistants are greatly appreciated.

Table 4 – Summary of Laboratory and Field Test Performance

Relative Performance of Oil Indicated[1]

Phase of Performance	Test Location	Oil Designation A	B	C	D	E
Bearing Corrosion	Laboratory	3	2	1	1	1
	Field	3	2	1	1	1
Ring Sticking	Laboratory	2	3	3	1	1
	Field	2	3	3	1	1
Oil Ring Deposits	Laboratory	3	2	2	1	1
	Field	2	2	2	1	
Air Port Deposits	Laboratory	2	3	1	1	1
	Field	1	1	1	1	1
Wear	Laboratory	No information available				
	Field	1	1	1	1	1
Piston Deposits	Laboratory	1	1	1	1	1
	Field	1	1	1	1	1
Purolator Deposits	Laboratory	2	2	3	1	1
	Field	2	2	3	1	1
AC Filter Deposits	Laboratory	1	1	1	1	1
	Field	1	2	2	1	1
Used Oil Analyses	Laboratory	1	1	1	1	1
	Field	1	1	1	1	1
Average	Laboratory	1.9	1.9	1.8	1	1
	Field	1.6	1.7	1.7	1	1

[1] 1 indicates best performance, 3 poorest performance.

Appendix I

Summary of Ring Sticking Results

Oil Designation	500-Hr Laboratory Test: Engine and Run No.	Number of Compression Rings Stuck	Average Number of Rings Stuck Per Engine	60,000-Mile Field Test: Coach No.	Number of Compression Rings Stuck	Average Number of Rings Stuck Per Engine
A	1-19	2	2	213	4[2]	4
				217	3[3]	
B	2-1	1[1]	3	210	13	9
B	2-3	4		214	5	
C	4-2	3	3	212		
				216	10	
D	1-18	0	0	211	0	
				215	3	
E	4-1	0	0	715	1	
				718	3	

[1] At 306 hr, test termination.
[2] At 37,602 miles, test termination.
[3] At 50,027 miles, test termination.

Appendix II

Summary of Oil Ring Deposit Results

Oil Designation	500-Hr Laboratory Test: Engine and Run No.	Oil Ring Clogging	60,000-Mile Field Test: Coach No.	Average Closure of Oil Ring Passages[2], %
A	1-19	Heavy	213, 217	50
B	2-1, 2-3	Moderate	210, 214	60
C	4-2	Moderate	212, 216	
D	1-18	Slight	211, 215	
E	4-1	Slight	715, 718	

[1] Many compression rings found broken, heavy oil ring deposits probably attributable to high blowby rate.
[2] Since rings were turned or renewed before test termination, clogging of each set of rings was plotted against accumulated mileage on that set and the curves extrapolated to 60,000 miles to yield the value reported.

Appendix III

Summary of Air Port Deposit Results

Oil Designation	500-Hr Laboratory Test: Engine and Run No.	Number of Air Port Cleanings	Average Number of Air Port Cleanings	60,000-Mile Field Test: Coach No.	Closure of Air Ports, %	Average Closure of Air Ports, %
A	1-19	1	1	213	0[2]	0
				217	0[3]	
B	2-1	2[1]	2	210	20	15
B	2-3	2		214	5	
C	4-2	0	0	212	5	10
				216	10	
D	1-18	0	0	211	25	25
				215	20	
E	4-1	0	0	715	15	15
				718	10	

[1] At 306 hr, test termination.
[2] At 37,602 miles, test termination.
[3] At 50,027 miles, test termination.

Appendix IV

Summary of Ring Wear Data – Field Tests

Oil Designation	Coach No.	Average Ring Weight Loss, g per cyl: Ring 1	Ring 2	Ring 3	Ring 4	Total (Rings 1-4)	Miles: Test	Liner
A	213	0.75	0.36	0.24	0.20	1.55	37,602	109,838
A	217	0.26	0.06	0.03	0.02	0.37	50,027	107,886
B	210	1.83	0.85	0.67	0.57	3.92	59,346	148,141
B	214	0.71	0.61	0.43	0.36	2.11[1]	58,968	58,968
C	212	1.90[2]	0.93	0.52	0.46	3.81	61,547	134,244
C	216	1.75	0.91	0.49	0.72	3.87	61,441	134,888
D	211	0.84	0.32	0.43	0.14	1.73	63,006	124,742
D	215	1.50	0.74	0.46	0.38	3.08	65,176	134,799
Average	....	1.19 (47%)	0.60 (23%)	0.41 (16%)	0.36 (14%)	2.56 (100%)		

[1] Abnormally high wear for indicated total liner mileage undoubtedly caused by timing gear failure, which terminated test, value not plotted in Fig. 6.
[2] Value estimated since all No. 1 rings broken and no actual weights available.
Note: No wear data are available on Oil E.

Appendix V

Summary of Piston Deposit Results

Oil Designation	500-Hr Laboratory Test: Engine and Run No.	Ring Groove Deposits	Piston Skirt Deposits	Deposit Above Top Ring	60,000-Mile Field Test: Coach No.	Ring Groove Deposits	Piston Skirt Deposits	Deposit Above Top Ring
A	1-19	Heavy	Medium, Black	Hard, Medium	213[1]	Normal	Medium, Black	Hard, Light to Medium
					217[2]	Normal	Medium, Black	Hard, Light to Medium
B	2-1[3]	Medium	Heavy, Black	Hard, Medium	210	Normal	Medium, Black	Hard, Light to Medium
B	2-3	Medium	Heavy, Black	Hard, Medium	214	Normal	Medium, Black	Hard, Light to Medium
C	4-2	Medium	Heavy, Black	Hard, Medium	212	Normal	Medium, Black	Hard, Medium
					216	Normal	Medium, Blac	Hard, Medium
D	1-18	Medium	Medium, Black	Hard, Medium	211	Normal	Medium, Black	Hard, Medium
					215	Normal	Medium, Black	Hard, Medium
E	4-1	Medium	Medium, Black	Hard, Medium	715	Normal	Medium, Black	Hard, Medium
					718	Normal	Medium, Black	Hard, Medium to Heavy

[1] At 37,602 miles, test termination.
[2] At 50,027 miles, test termination.
[3] At 306 hr; test termination.
Note: Under-piston deposits were negligible in all cases.

Appendix VI

Summary of Purolator Deposit Results

Oil Designation	Engine and Run No.	Number of Purolator Filter Changes	Average Number of Purolator Filter Changes	Coach No.	Average Purolator Deposits[1]	Average Purolator Deposits[1]—Both Coaches
A	1-19	2	2	213	L-M[2]	T-L
				217	C-T[3]	
B	2-1	1[4]	1	210	L	L
B	2-3	1		214	L	
C	4-2	3	3	212	M	M
				216	M	
D	1-18	0	0	211	T	T
				215	T-L	
E	4-1	1	1	715	C-T	T
				718	T-L	

[1] Ratings: C = clean, T = trace, L = light, M = medium, H = heavy, VH = very heavy.
[2] Test terminated at 37,602 miles.
[3] Test terminated at 50,027 miles.
[4] Test terminated at 306 hr.

Appendix VII

Summary of AC Filter Deposit Results

Oil Designation	Engine and Run No.	Number of AC Filter Changes	Average Number of AC Filter Changes	Coach No.	Average AC Filter Deposits[1]	Average AC Filter Deposits[1]—Both Coaches
A	1-19	2	2	213	L[2]	C-L
				217	C[3]	
B	2-1	2[4]	2	210	H-VH	H
B	2-3	1		214	H	
C	4-2	2	2	212	H	H
				216	M-H	
D	1-18	2	2	211	C	C-L
				215	L-M	
E	4-1	2	2	715	L	L-M
				716	M	

[1] Ratings: C = clean, L = light, M = medium, H = heavy, VH = very heavy.
[2] Test terminated at 37,602 miles.
[3] Test terminated at 50,027 miles.
[4] Test terminated at 306 hr.

Appendix VIII

Summary of Used Oil Examinations

	500-Hr Laboratory Test					60,000-Mile Field Test				
		Average[1] Tests on Used Oil, Increase over Original Values					Average Tests on Used Oil at 10,000-Mile Periods, Increase over Original Values			
				Sludge, mg per 10g oil					Sludge, mg per 10g oil	
Oil Designation	Engine and Run No.	Viscosity, SU at 100 F	Neutralization No.	Dissolved[2]	Undissolved[3]	Coach No.	Viscosity, SU at 100 F	Neutralization No.	Dissolved[2]	Undissolved[3]
A	1-19	6	0.01	41	36	213	−94	0.39	39	87
						217	−47	0.45	49	3
B	2-1	93	1.08	193	84	210	42	0.59	37	104
B	2-3	38	0.14	46	14	214	5	0.53	50	108
C	4-2	50	−0.66	16	13	212	55	−0.07	10	108
						216	27	−0.07	24	146
D	1-18	10	0.06	28	22	211	−70	0.17	20	42
						215	22	0.03	16	73
E	4-1	18	−0.33	20	13	715	20	−0.02	18	56
						718	77	0.05	19	145

[1] Average value after AC filter installed and oil conditions stabilized.
[2] See Industrial and Engineering Chemistry, Analytical Edition, Vol. 11, April 15, 1939, pp. 183-185: "Determination of Dissolved Sludge in Used Oils," by Frank W. Hall, Harry Levin, and Wallace A. McMillan.
[3] See Industrial and Engineering Chemistry, Analytical Edition, Vol. 11, April 15, 1939, pp. 181-183: "Determination of Undissolved Sludge in Used Oils," by Harry Levin and Charles C. Towne.
Note: Dilution caused a decrease in viscosity in many cases.

Laboratory Testing of Heavy-Duty Oils

OUR early research on the problem of heavy-duty motor oil development concerned the actual temperature measurement of the hotter parts of the piston to which a lubricant would be exposed. This information was necessary to furnish a background of actual engine temperature conditions accountable for lubricant decomposition, particularly in the upper ring belt area where such decomposition caused ultimate engine failure by giving rise to piston-ring sticking. Search for a single-cylinder laboratory test engine possessing flexibility and conservative costs demanded of laboratory equipment centered on the Fairbanks-Morse model 36A diesel engine. This engine has a 4¼-in. bore and 6-in. stroke and is rated at 10 hp.

Various combinations of temperature measuring devices were studied in an attempt to design the most efficient system for measurement of piston temperatures. A special jig for holding the piston was designed in order that thermocouple wells could be accurately located at any point on the piston. Iron-constantan thermocouples were constructed using high-melting-point silver solder. The thermocouples were silver soldered to individual iron-constantan contactors of the piston-plate assembly. Thermocouple wires were led from contactor fingers through the side inspection plate to a sensitive millivoltmeter. Piston temperatures were measured by the make-and-break action of the contactors of the piston-plate assembly with the contact fingers.

Various Test Conditions Studied

Establishment of an accelerated laboratory engine test to evaluate ring-sticking characteristics of crankcase lubricants necessitated considerable experimentation involving variations of jacket temperature, loads, and speeds. The ultimate goal was the establishment of a given set of test conditions that could be duplicated within reasonable limits and still evaluate different lubricants in the order of field service performance. Many tests involving two reference oils, one poor and the other satisfactory in field operation, showed that consistent ring-sticking times and overall cleanliness comparisons consistent with field performance could be obtained under the following test conditions:

Type of piston	Aluminum
Bearings	Babbitt
Load, hp	9.3 (34-lb beam load)
Speed, rpm	1200
Jacket temperature, F	350 (ethylene glycol coolant)
Oil volume, cc	3800 (no make-up).

Generally speaking, correlation of ring sticking with increased temperature rise of the piston-ring area was successful for about 40% of the oils evaluated. This inconsistency was no doubt due to build-up of insulating carbon deposits on the piston thermocouples, which in turn rendered the temperature measuring devices less sensitive to temperature changes at the time of ring sticking. Of the many oils evaluated, the loss in power output was found to be more sensitive to sluggish ring action or ring sticking than increased piston temperatures.

Oils of the Gulf Coastal type, while prolonging ring-sticking time, produced extremely dirty pistons, badly stuck rings, and overall engine sludge that was difficult to remove. This cleanliness factor was, therefore, another lubricant characterisitc to be considered in evaluating and producing heavy-duty type lubricants.

Installation of automatic recording blowby equipment and crankcase sealing was later provided which simplified the operating detail of the engine in evaluating ring-sticking characteristics of the heavy-duty types of lubricating oil or the mineral oil base stocks used for blending. Substitution by the blowby recording mechanism was successful in predicting both faulty ring action and ultimate stuck ring conditions of all oils evaluated.

This early work on piston operating temperatures and ring-sticking characteristics for various type base oils was sufficient to allow us to alter the test procedure and equipment in order that we might better evaluate base oils containing additives. The first change made was to use a cast-iron piston in place of the aluminum piston previously used. It was felt that the use of the cast-iron piston would provide a more severe test and would last longer than the aluminum piston. Because of the increased severity of the test due to the piston change we were obliged to reduce the cooling jacket temperature from 350 F to 325 F in order to obtain sufficient differences between various heavy-duty type oils. Shortly after this change, it became apparent that an evaluation to supplement ring-sticking time was in order and piston cleanliness was chosen as the additional evaluation. A running time of 24 hr per test was then set up and the oils evaluated for both piston cleanliness and ring-sticking characteristics. The test procedure as now set up requires a 24-hr run regardless of whether or not rings are stuck prior to that time as may be indicated by the automatic blowby recorder.

The piston is rated by the merit system, which gives a rating of 5 for a clean piston with no stuck rings and a rating of 1 for a dirty piston with stuck rings. This engine test procedure now allows us to evaluate additive oils as well as base oils.

Standard Test Conditions

Standard conditions for all runs now are as follows:

Speed, rpm	1200
Load, hp	9.3
Jacket Temperature, F	325 (ethylene glycol coolant)
Oil Sump Temperature, F	230-240 (not controlled)
Volume of crankcase oil, cc	3800 (initial volume)

One quart make-up added at 10 and 20 hr and excess oil drained to maintain 3800-cc level.

Our experience has shown that a 24-hr run on the Fairbanks-Morse engine with a piston rating of 3.5 or better with no stuck rings will give a satisfactory 480-hr Caterpillar test and consequently saves much costly engine time.

Excerpts from the paper: "The Fairbanks-Morse Diesel Engine as a Research Tool for Pre-Evaluation of Heavy Duty Motor Oils," by H. L. Moir, W. J. Backoff, and N. D. Williams, The Pure Oil Co., presented at the SAE Diesel Engine and Fuels & Lubricants Meeting, Cleveland, June 2, 1943.

ther
and

nent
ified
tick-
g oil
stitu-
al in
ring

and
was
quip-
con-
cast-
ously
vould
n the
of the
educe
order
-duty
t that
as in
tional
en set
s and
now
or not
ed by

ives a
and a
engine
oils as

vs:

olant)
)

cess oil

e Fair-
better
aterpil-
ime.

Diesel-
Heavy-
N. D.
Diesel-
l, June

No. 12

29 V

W
530

917
R.

29
No

29
C.
E
29
A.
3-2

The
or
at